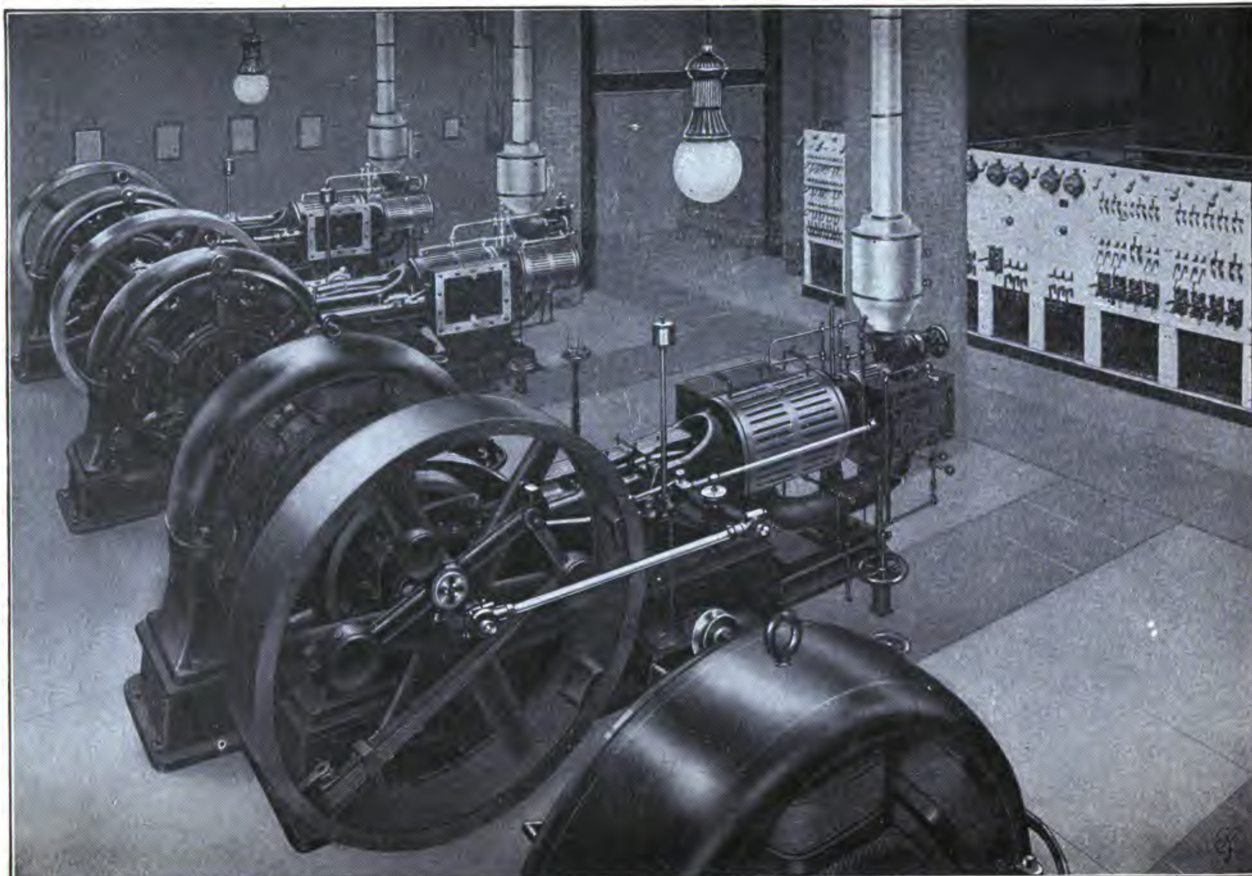
STEAM ENGINES

LUDY

AMERICAN SCHOOL OF CORRESPONDENCE
CHICAGO ILLINOIS



FOUR 200-H. P. TANDEM-COMPOUND AUTOMATIC ENGINES, DIRECT-CONNECTED TO GENERATOR
Skinner Engine Co., Erie, Penna.

STEAM ENGINES

A THOROUGH AND PRACTICAL PRESENTATION OF MODERN STEAM ENGINE PRACTICE

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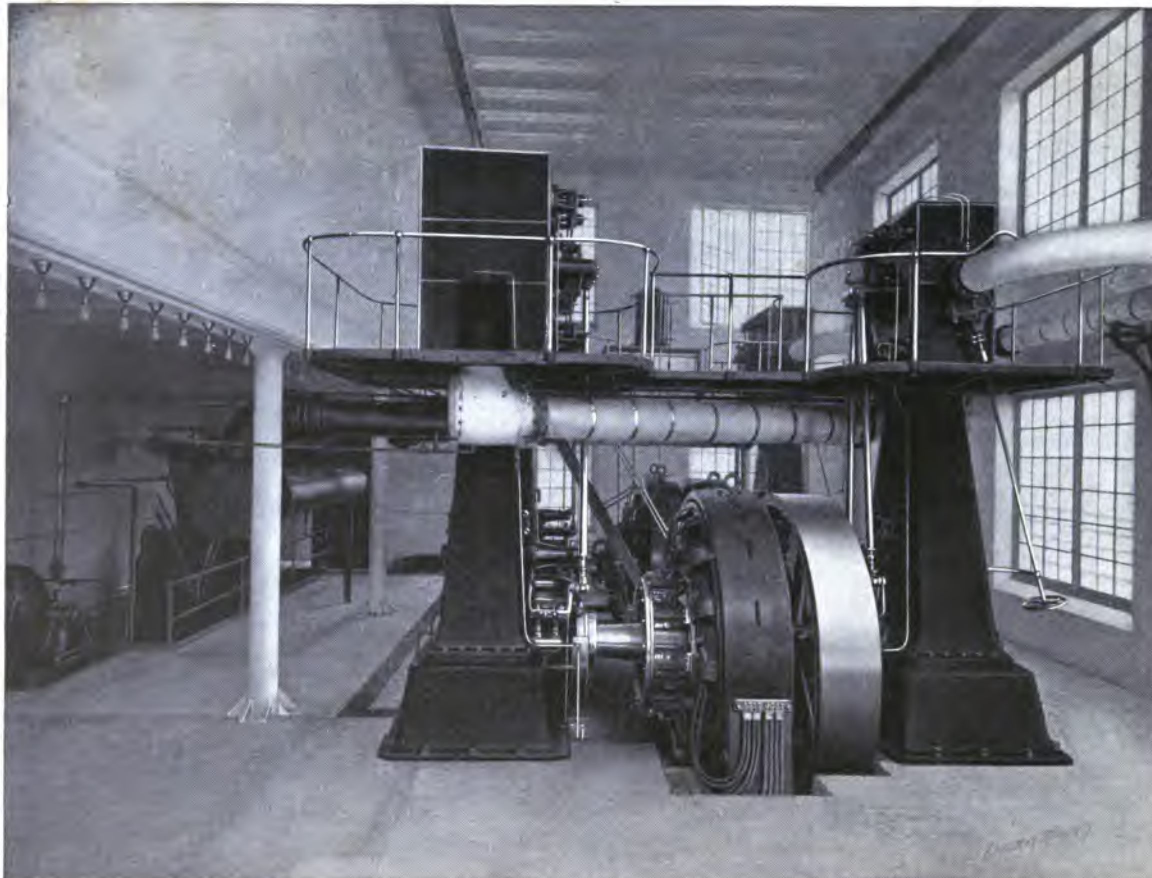


VERTICAL CROSS-COMPOUND CORLISS ENGINE IN THE NEW YORK AND PENNSYLVANIA RAILWAY PLANT AT JOHNSONBURG, PENNA.
Ball Engine Co., Erie, Penna.

INTRODUCTION

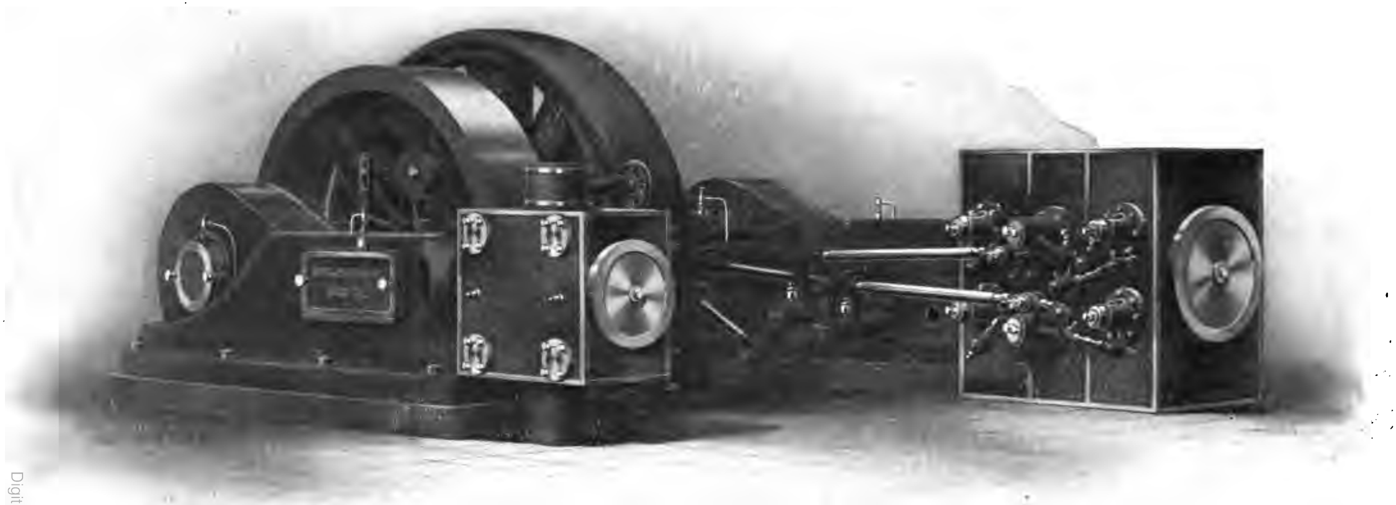
THE modern steam engine, whether it be the majestic Corliss, which so silently operates the massive electric generators in one of our municipal power plants, or the giant locomotive which pulls the Limited at sixty miles an hour, commands our unstinted admiration. And yet every movement is so free and perfect in its action, every function is performed with such precision and regularity, that we lose sight of the wonderful theoretical and mechanical development which was necessary to bring these machines to their present state of perfection.

¶ The genius of Watt, the "father" of the steam engine, was so great that his basic conception of this, his greatest invention, and of many of his minor discoveries in connection with it, remain almost as he gave them to the world over a century ago. Yet he was so far in advance of the mechanical development of his time that his workmen could not build engine cylinders nearer true than three-eighths of an inch. Modern builders demand an accuracy of at least two-thousandths of an inch—almost two hundred times greater.

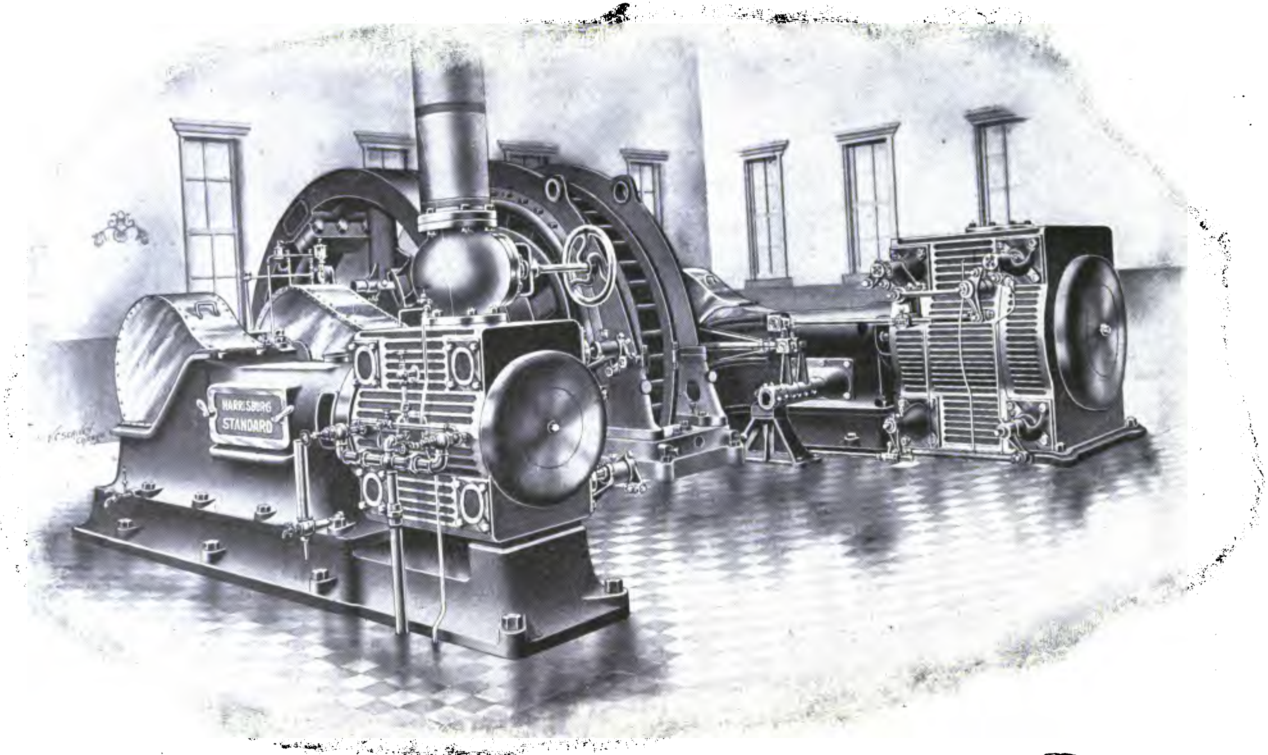
¶ But mechanical skill is not the only particular in which progress had been made. Many minor but important improvements have been brought about by a careful study of the theory of heat engines; the reduction of enormous heat losses, the use of superheated steam, the idea of compound expansion, the development of the Stephenson, Walschaert, and other valve gears—all have contributed towards making the steam engine well-nigh mechanically perfect and as efficient as is inherently possible.

¶ This story has been developed from a historical standpoint and along sound theoretical and practical lines. It will be found

absorbingly interesting and instructive to the stationary engineer as well as to all who wish to follow modern steam engineering development. The material is particularly adapted to home study as it has been prepared for use in the American School's regular correspondence courses. If, therefore, the book should prove of real value in stimulating the interest of the trained man or the layman in the technical developments of the day, the publishers will feel that its mission has been accomplished.



HORIZONTAL CROSS-COMPOUND SIDE-CRANK CORLISS-TYPE STEAM ENGINE
The Ball Engine Co., Erie, Penna.



GENERAL CONSTRUCTION FLEMING CROSS COMPOUND FOUR-VALVE ENGINE DIRECT-CONNECTED STYLE, GENERAL VIEW,

Harrisburg Foundry and Machine Works.

STEAM ENGINES

PART I

DEVELOPMENT

Early History. In the study of this subject, it is thought advisable to review the historical development of the steam engine in order that a broad conception of it may be obtained. It is not intended, however, to give the history of the steam engine in detail—although it is an exceedingly interesting one, which would be beneficial for any one to review—but rather, a short résumé in order that the student may be prepared for a detailed study of the modern engine.

The first steam engines of which we have any knowledge were described by Hero of Alexandria, in a book written two centuries before Christ. Some of them were very ingenious, but the best were little more than toys. From the time of Hero until the seventeenth century little progress was made. At this time, however, there was a great need of steam pumps to remove water from the coal mines. In 1615, Salomon de Caus devised an arrangement, consisting of a vessel having a pipe leading from the bottom which was filled with water and then closed. When heat was applied to the vessel, steam was formed, which forced the water through the discharge pipe.

Later an engine was constructed in the form of a steam turbine, but was unsuccessful, and the attention of the inventors was again turned to pumps.

Savery. Finally Thomas Savery completed, in 1693, the first commercially successful steam engine. It was very wasteful of steam as compared with engines of today but, as being the first engine to accomplish its task, it was successful. Savery's engine, Fig. 1, consisted of two oval vessels A_1 and A_2, placed side by side and in communication with a boiler B_1. The lower parts were con-

nected by tubes fitted with suitable valves. In operation, steam from the boiler was admitted, say, to the vessel A_2 and the air driven out. The steam was then condensed and a vacuum formed by letting water play over the surface of the vessel. When valve *1* was opened, this vacuum drew water from below until the vessel was full. The valve was then closed and steam again admitted by valve *2*, so that on opening valve *3* the water was forced out through the delivery pipe *C*. The two vessels worked alternately. When one was filling with water, the other was open to the boiler and was being emptied. Of the two boilers B_1 and B_2, one supplied steam to the oval vessels and the other was used for feeding water

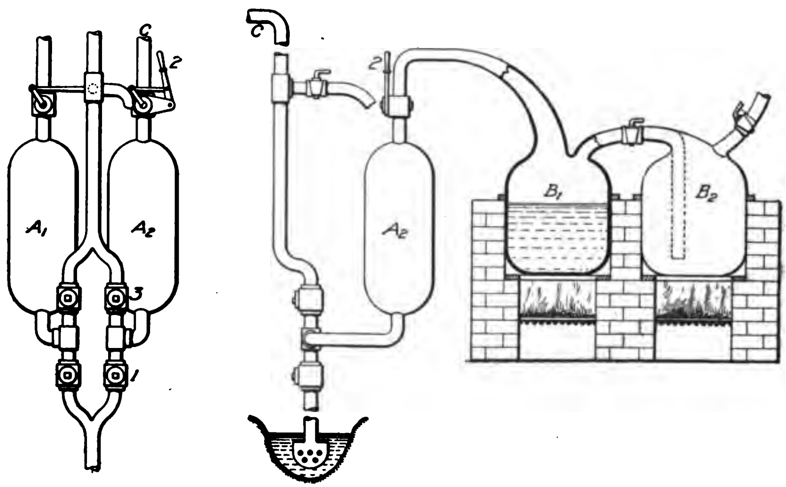


Fig. 1. Early Form of Steam Pumping Engine

to the first boiler. In operation the second boiler was filled while cold, and after a fire had been lighted under it, acted like the vessel used by Salomon de Caus and forced a supply of feed water into the main boiler.

A modification of Savery's engine—the pulsometer shown in Fig. 2—is still found in use in places where an ordinary pump could not be used and where extreme simplicity is of especial advantage. Its valves work automatically and it requires very little attention.

A serious difficulty with Savery's engine resulted from the fact that the height to which water could be raised was limited by the pressure which the vessels could sustain. Where the mine was

very deep it was necessary to use several engines, each one raising the water a part of the whole distance. The consumption of coal in proportion to the work done was about twenty times as great as that of a good modern steam engine. This was largely, though not entirely, due to the immense amount of steam which was wasted by condensation when it came in contact with the water in the oval vessels.

Newcomen. The next great step in the development of the steam engine was taken by Newcomen, who in 1705 succeeded in developing a scheme which prevented contact between the steam and the water to be pumped, thus diminishing the amount of steam uselessly condensed. He introduced the first successful engine which used a piston working in a cylinder.

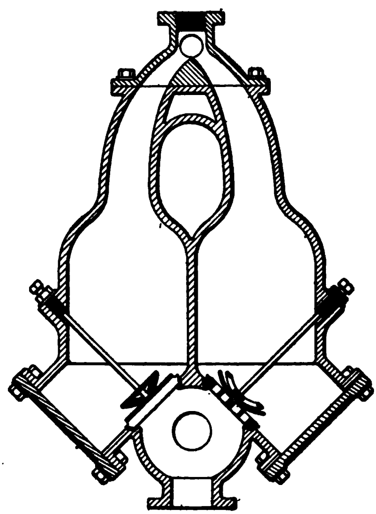


Fig. 2. Pulsometer

In Newcomen's engine, Fig. 3, there was a horizontal lever *A*, pivoted at the center, carrying at one end a long heavy rod *B* which connected with a pump in the mine below. A piston *C* was hung from the other end of the lever and worked up and down in a vertical cylinder *D*, which was open at the top. Steam acting on the lower side of the piston, at atmospheric pressure, was admitted from the boiler to the cylinder, and as the pressure was the same both above and below the piston, the weight of the heavy pump rod raised the piston. A jet of water in the cylinder condensed the steam and formed a vacuum. This left the piston with atmospheric pressure above and very little pressure below (a partial vacuum), so it was forced down and the pump rod raised again. Steam could again be admitted to the cylinder; the pump rod would fall; and the process could be continued indefinitely.

In the days of Newcomen it was very difficult to obtain good workmanship. For this reason it was often necessary to make the cylinders of wood. In order to prevent steam from blowing

around the piston, or air from leaking in where steam was being condensed, it was customary to keep a jet of water playing on the top of the piston.

One great trouble with all of these engines was that some one was required to open and close the cocks, and boys were generally employed to do this work. One boy, in order to get time to play, rigged a catch at the end of a cord which was attached to the beam

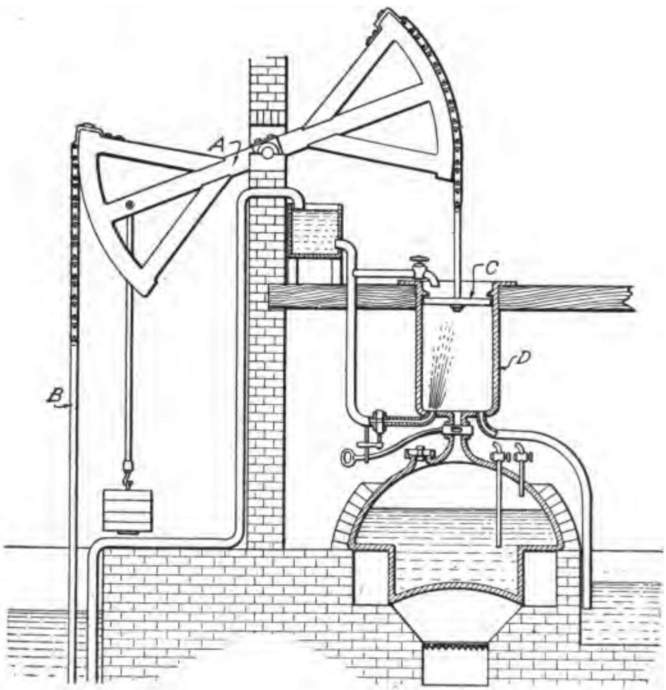


Fig. 3. Newcomen's Steam Pumping Engine

overhead, and this did the work for him. By making the valves in this way automatic, made it possible to dispense with the services of the boy and at the same time greatly increase the speed of the engine.

The Newcomen engine was improved slightly from time to time by different inventors and was very extensively used until the time of Watt, a very few of them still being in existence today. While this engine was a success and a great improvement over its

predecessors, it was still very large, wasteful, and heavy in comparison with the work done, and the cylinders, when made of iron, were simply cast and not bored, thus leaving a rough, inner wall.

Watt. In the year 1763, a small model of a Newcomen engine was taken to the shop of an instrument maker in Glasgow, Scotland, to be repaired. This instrument maker, whose name was James Watt, had been studying steam engines for some time and he became very interested in this model. He was a man of great genius, and before he died his inventions had made the steam engine so perfect a machine that there has been but one really great improvement in it since his time, namely, compounding.

He found that to obtain the best results it was necessary, "*first,* that the temperature of the cylinder should always be the same as that of the steam which entered it; and *second,* that when steam was condensed it should be cooled to as low a temperature as possible." All improvements in steam-engine efficiency have been in the direction of a more complete realization of these two conditions.

In order to keep the cylinder nearly as hot as the entering steam, Watt no longer injected water into the cylinder to condense the steam, but used a separate vessel or condenser. He made his piston tight by using greater care in construction, so that it was unnecessary to have a water seal at the top. He then covered the top of the cylinder to prevent air from cooling the piston. When this was done he could use steam above the piston as well as below; this made the engine double acting.

Also, in the effort to keep the cylinder as hot as the entering steam, he enclosed the cylinder in a larger one and filled the space between with steam. This was not often done, however, and only of late years has the steam jacket been of much advantage. Also, the steam was used expansively, that is, the admission of steam was stopped when the piston had made a part of its stroke; the rest of the stroke was completed by the expansion of the steam already admitted. This plan is now used in all engines that are built for economy.

Other inventions made by Watt on his steam engines were: a *parallel motion,* that is, an arrangement of links connecting the end of the piston rod with the beam of the engine in such a way as to guide the rod almost exactly in a straight line; a *throttle valve* for

regulating the rate of admission of steam; and a *centrifugal governor*, which controlled the speed of the engine shaft by acting on the throttle valve. Watt's engine as finally developed is shown in Fig. 4.

Watt saw that by using high-pressure steam he could get more work from it; but as it was not possible to make a very reliable

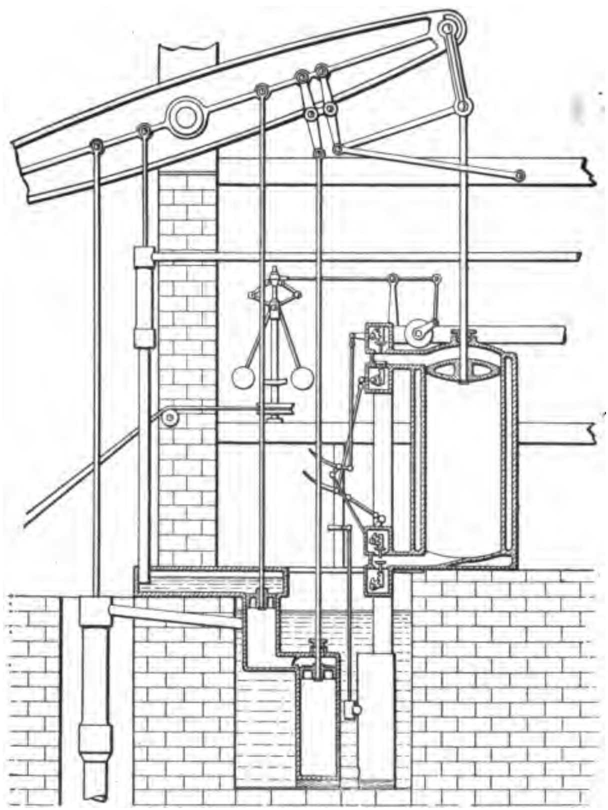


Fig. 4. Final Form of Watt's Steam Pumping Engine

boiler he never used a pressure of more than seven pounds per square inch above the atmosphere. About the year 1800, comparatively high pressures came more into use and the non-condensing engine was introduced. In Watt's engine, and all those preceding his, a vacuum was produced in front of the piston by condensing the steam, and either the atmosphere or steam at atmospheric pressure pushed

it through the stroke. In the non-condensing engine, using high-pressure steam, the space in front of the piston could be opened to the atmosphere at exhaust and, although the atmospheric pressure resisted its motion, the pressure of the steam behind the piston was still greater than that of the air. These engines were much more simple than the condensing engines, as they required no condenser.

Compound Pumping Engine. About this time what would now be called a compound engine was introduced by Hornblower and later by Woolf. It had two cylinders of different size, steam being admitted into the smaller one and then passing over into the larger. Only a little expansion occurred in the small cylinder and much more in the larger one.

About the year 1814, Woolf introduced a compound pumping engine in the mines of Cornwall, but a simpler engine was later introduced and Woolf's engine fell into disuse. This later engine became known as the *Cornish pumping engine* and was famous for many years because of its economy. It was the first engine ever built that could compare at all with modern engines in the matter of steam consumption. It consisted of a single cylinder placed under one end of a beam from the other end of which hung a heavy rod which operated a pump at the foot of the shaft. Steam was admitted to the upper side of the piston for a short portion of the stroke and allowed to expand for the remainder of the stroke. This forced the piston down, lifted the heavy pump rod, and filled the pumps with water. Then communication was established between the upper and under side of the piston, exhaust occurred, and the heavy pump rod fell, lifting the piston and forcing the water out of the pumps. Steam was cut off at about three-tenths stroke, and the pump made about seven or eight complete strokes per minute with a short pause at the end of each stroke to allow the valves to close easily and the pumps to fill with water. These engines needed great care and were in charge of competent men, to whom prizes were frequently given for the best efficiency, which doubtless accounts for their wonderful performance.

Parts of Steam Engine. Leaving the historical side of the steam engine let us now turn to the modern simple steam engine and study briefly its construction. Figs. 5, 6, and 7, will serve to illustrate a

horizontal, center crank engine, all the more important parts being numbered. The function of the various parts will be considered in detail later in the work.

Referring to the numbers in Figs. 5, 6, and 7, the names of the parts are shown in the following list:

List of Parts

Sub-base *1*
Frame *2*
Main bearing caps *3*
Main bearing liners *4*
Cylinder *5*
Cylinder head *6*
False head cover *7*
Valve chest head (head end) *8*
Valve chest head (crank end) *9*
Piston *10*
Piston rings *11*
Piston rod *12*
Piston rod nut (piston end) *13*
Piston rod nut (crosshead end) *14*
Piston rod stuffing box *15*
Piston rod gland *16*
Crosshead *17*
Crosshead shoes *18*
Crosshead adjusting screws *19*
Crosshead pin *20*
Crosshead pin nut *21*
Cross pin washer *22*
Connecting rod *23*
Connecting rod bolts *24*
Connecting rod strap *25*
Crosshead pin box *26*
Crosshead pin box wedge *27*
Adjusting screws *28*
Crank pin box *29*
Crank pin box wedge *30*
Adjusting screws *31*
Crank disks *32*
Crank shaft *33*
Flywheels *34*
Valve pistons *35*
Valve rings *36*
Valve cages *37*
Valve rod *38*
Valve rod nuts (valve end) *39*
Valve rod nuts (ram end) *40*
Valve rod gland *41*
Ram box *42*
Ram box caps *43*
Ram *44*
Ram pin and nut *45*
Ram pin cap *46*
Eccentric rod connection *47*
Eccentric rod *48*
Eccentric rod nut (ram end) *49*
Eccentric rod nut (eccentric end) *50*
Eccentric *51*
Eccentric strap *52*
Dash plate *53*
Dash plate gland *54*
Doors *55*
Door handle *56*
Door clamps *57*
Oil hood *58*
Oil hood handles *59*
Eccentric oil boat *60*
Valve rod oil boat *61*
Oil vent *62*
Sheet steel lagging *63*
Drain cocks *64*
Shaft governor *65*

Sub-Base. The sub-base *1*, Fig. 6, is made of a good grade of cast iron and is usually heavily ribbed and made high enough to permit the wheels to clear the floor. The sub-base is often omitted with engines of large size, the engine being set upon a concrete base.

Frame. The frame *2* is the element or link by which all of the parts of the engine are held in place, so that their relative positions

are always maintained to the end that their proper functions may be performed. The frame is a heavy, substantial casting so designed

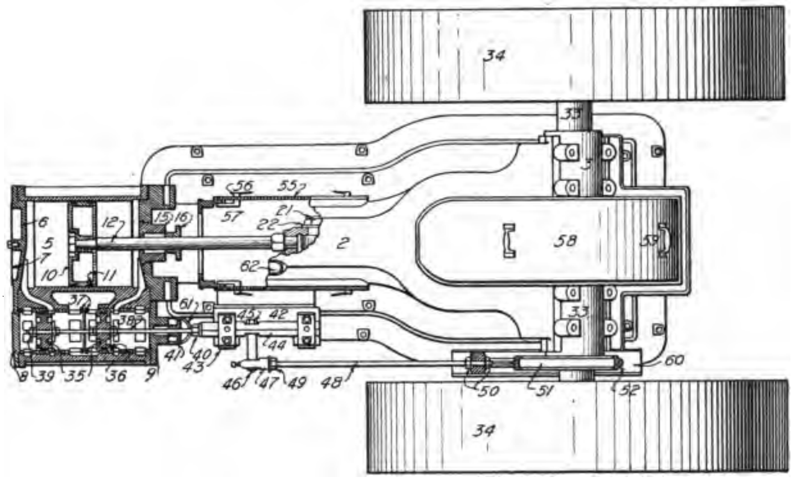


Fig. 5. Plan View of Modern Simple Engine

that it is strong enough to take all the stresses put upon it. The type, size, and details of the frame vary with the type and size of

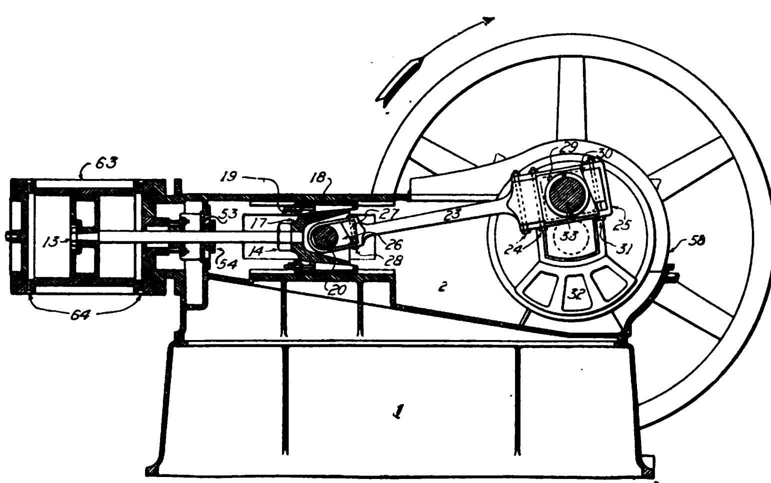


Fig. 6. Side Elevation of Modern Simple Engine

the engine of which it is a part. Usually the lower guide, valve rod guide, and seats for the main bearings are cast integral with it. In

small sizes the cylinder is frequently cast integral with the frame. Provision is always made for adjustments necessitated by any wear of the frame or parts attached thereto. It is to be noted in Fig. 6, that the frame crank case *2* is connected with the crosshead guide. It frequently has an opening into the sub-base, thus permitting the oil from the crosshead, guides, and crank to drain into a suitable receptacle in the sub-base, from which it is taken by means of a drain cock conveniently located in the side or end. The crank is enclosed

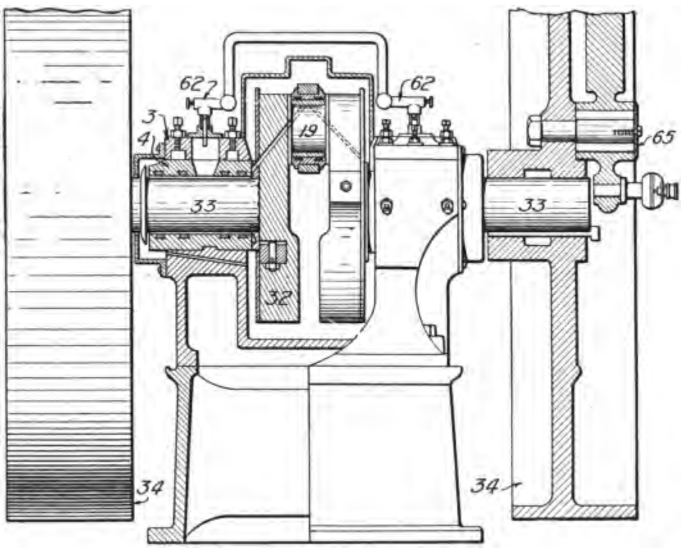


Fig. 7. End Elevation and Part Section of Modern Simple Engine

by a neat, pressed, sheet steel cover, which prevents the oil from being thrown outward on the floor while the engine is running. Quite frequently the crank cover is made of cast iron.

Cylinders. The cylinder *5*, Fig. 5, is one of the most important parts of the steam engine, for it is in the cylinder that the energy of the steam is converted into useful work. The cylinder is circular in section and is attached to the bed by means of a number of bolts. It is made of close grain, gray cast iron. The casting of the cylinder should be done with great care, so as to insure a casting free of blow-holes or other defects.

Fig. 8 illustrates the cylinder in cross section as well as showing its contained parts. The cylinder barrel *1* is accurately bored

and fitted. Inside of this barrel the piston *2* is driven back and forth by the steam, which is admitted alternately on one side and then on the other through the ports *13*. The piston is connected to the crosshead through the piston rod *3*. The continuous movement back and forth of the piston causes the surface of the cylinder to wear away, and in order to avoid a shoulder being formed by this action, the cylinder is counterbored at each end by an amount depending on the size of the cylinder. The diameter of the counter-

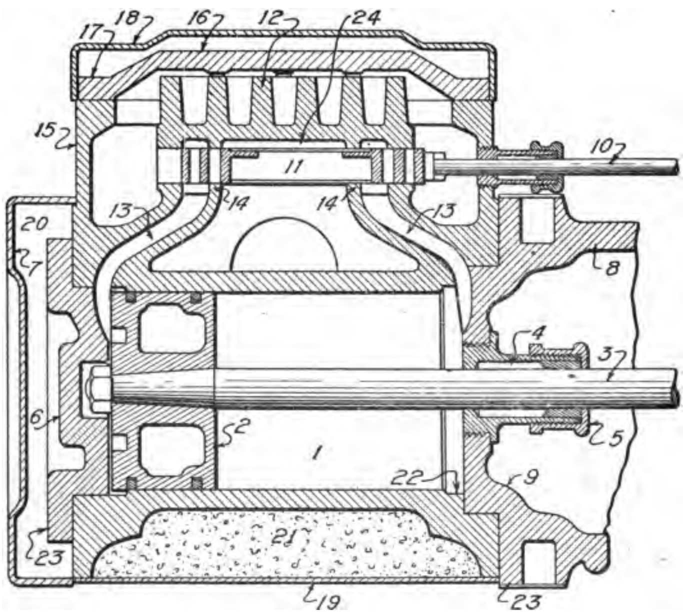


Fig. 8. Cylinder and Valve Mechanism Shown in Section

bore *22* is usually about one-quarter of an inch larger than that of the cylinder proper, depending, however, somewhat on the size of the cylinder. The stroke of the piston is such that the piston moves beyond the wearing surface at each stroke, thus preventing any shoulder being developed in the cylinder wall.

The cylinder is attached to the bed of the engine by a number of bolts which are placed through the flanges *23* of the cylinder and cylinder head. Each end of the cylinder is closed by means of the cylinder heads *6* and *9*. The cylinder head *6* is called the back cylin-

der head (head end), and *9* is known as the front cylinder head (crank end). In the illustration the front head *9* is a portion of the frame, but in many constructions it is entirely independent of the frame. In order to have a steam-tight cylinder, it is necessary to make a tight joint between the cylinder heads and the cylinder barrel. This is accomplished by turning both surfaces true, then grinding the joints with emery and oil. After the joints are well ground the heads are tightly drawn up against the cylinder by means of bolts suitably arranged. A sheet iron jacket *19* is put around the cylinder, leaving an air space *21* between the cylinder walls and the jacket. This air space retards the cooling off of the cylinder walls, hence initial condensation of the steam in the cylinder is reduced. In some types of engines such as locomotives, this air space is filled with some non-conducting material such as asbestos. This is also

Fig. 9. Piston, Showing Piston Rings for Making Steam-Tight Joints

sometimes done by builders of stationary engines. It should be noted also, that the back cylinder head has an air space for the same reason as that given for the space surrounding the cylinder. Since condensation does take place in the cylinder, some means must be provided for removing the water, hence the drain cocks *64*, Fig. 6, are placed in the bottom of the cylinder at each end. The pipe connection for these cocks enters the cylinder in the counterbore near the wearing surface. Any water that may be in the cylinder will be forced out through these cocks if they are open. Care must be taken that the cylinder is freed of water, for if it is not, on account of the incompressibility of water, the cylinder head may be forced off or other damage result therefrom. Some cylinders are provided

with relief valves, which automatically open when the pressure from any cause reaches a certain amount, thus preventing the bursting of a cylinder head.

Piston Rings. Between the piston *2*, Fig. 8, and the walls of the cylinder there must be a steam-tight joint, so that the live steam can not pass around the piston and be exhausted before expanded, otherwise a great waste of power will be incurred. The requirement is fulfilled by having the piston grooved, as shown in Fig. 9, and fitted with packing rings. These packing rings, commonly called *snap rings,* are turned up slightly larger in diameter than the cylinder and being cut, as shown in Fig. 9, they spring out into the cylinder, always pressing against the walls and forming an almost perfect steam joint. The piston of every engine is made with two or more of these packing rings. The cuts in the rings must not be placed directly in line with each other, otherwise the steam would have a better chance to blow through. In order to prevent this the joints are always placed on opposite sides of the piston. Packing rings are always made of cast iron, and are usually turned up to a uniform section. The outside portion and the two sides are carefully machined.

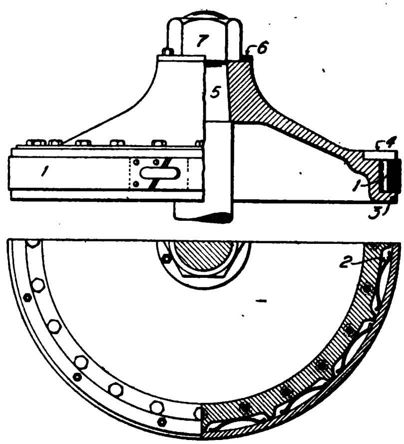


Fig. 10. Part Section Plan and Elevation of Conical Piston

Pistons. The piston *2*, Fig. 8, is usually made of cast iron, but sometimes is made of cast steel. It may be a solid disk grooved, as in Fig. 9, or the central portion may be cored out, as in Fig. 8.

Another type of piston that is largely used in marine and sometimes locomotive service is illustrated in Fig. 10. This is a comparatively light cast steel piston, but at the same time a very strong one, due to its conical construction. It will be noted also that only the one packing ring *1* is used. This packing ring is much wider than the ordinary snap ring and is pressed out against the cylinder wall by a number of single leaf springs being placed between the body of the piston and packing ring, as shown in Fig. 10 at *2*. The

piston is made with an L-shaped edge *3*, a band *4* being bolted on the open side of the L, thus forming a groove or opening for the reception of the small springs and the packing ring. The connection of the piston rod to the piston is also clearly shown. The rod *5* has a tapered end which is forced by hydraulic pressure into a tapered hole in the piston; the nut *7* is then tightened up and locked by placing the plate in position around the nut and fastening it with cap screws. This arrangement insures a lasting connection between the piston and the piston rod.

It is essential that the piston be as light as possible in order to reduce the amount of work absorbed in pulling it to and fro, and also to reduce the wear on the lower portion of the cylinder.

The piston rod *3*, Fig. 9, is fitted into a tapered hole in the piston and secured by means of a lock nut and cotter pin placed on the back end. Oftentimes the tapered fit is made very tight and the piston forced on by hydraulic pressure. An older form of attaching the piston rod to the piston is shown in Fig. 8. In this instance the rod has a tapered end, which is driven into a tapered hole in the piston where it is secured by a nut, no cotter pin being used. The other end of the piston rod is threaded, screwed into the crosshead, as shown in Fig. 6, and secured by means of a lock nut. In some constructions the crosshead end of the piston rod is tapered and secured by a key. Many schemes have been employed by different manufacturers for fastening the piston rod to the crosshead, all of which have their advantages and disadvantages. The piston rod, although usually made of a good quality of open hearth steel, is frequently made of nickel steel, which possesses great strength.

Stuffing Box and Packing. As the piston rod passes through the front cylinder head, some provision must be made for making a steam-tight joint between the piston rod and the cylinder head. This is accomplished by means of the stuffing box *4*, and the gland *5*, shown in Fig. 8. Some form of packing is placed around the piston rod within the stuffing box *4* and the gland is forced in by means of bolts or a secured cap as shown, thus holding the packing in the box and at the same time crowding the packing tightly against the piston rod.

The piston packing may be made of woven strands of hemp or cotton; or asbestos may be used. To insure lubrication of the rod

this fibrous packing is soaked in oil before being placed in position. In addition to this form of packing there are different compositions of rubber, graphite, cotton, etc., also various kinds of metallic packing in use. A metallic packing is made of material such as babbitt metal, which is a soft alloy of copper, tin, and antimony. This and other compositions are used for metallic packing, and the metal, being comparatively soft, wears away much more rapidly than that of the piston rod. Fig. 11 illustrates one form of packing, known as the U. S. Metallic packing. The principle of operation is as follows: The babbitt metal rings *2*, consisting of three rings cut in half, provide the packing and are the only parts which come in contact with the rod. These rings are forced into the vibrating cup *6* against the rod, and are fed down as wear takes place by the pressure of the steam itself. The spring behind the follower *3* is merely intended to hold the rings and other parts in place when steam is shut off. A ground joint is made between the flat faces of the vibrating cup *6* and the ball joint *4*. There is also a ground joint between the ball joint *4* and the gland *7*. The combination of the sliding face of the vibrating cup and the ball joint permits the packing to follow the rod freely without any increase in friction should it run out of line for any cause. This is an important feature, since the wear of the crosshead, guides, piston head, and cylinder produces an irregular alignment of the piston rod, which would injure the packing to a marked degree, if it was not flexible. The parts of the packing are held in place by the gland *7*, which is bolted to the cylinder head. A steam-tight joint is made between the gland and the cylinder head by means of a copper gasket. The purpose of the swab cup *5* is to hold in place a swab, which is usually made of waste, candle wicking, or a braided material, soaked in oil and oiled from time to time as a means of keeping the piston rod well

Fig. 11. Stuffing Box Packed with Metallic Packing

lubricated. In addition to this service, the swab catches and retains a considerable amount of dust and grit which would otherwise find its way into the cylinder, where it might do harm. It is to be said in favor of the various so-called rubber packings, that they give very good service. The four different styles of rubber packing illustrated in Fig. 12, are not composed entirely of rubber, but contain other material such as graphite, cotton, etc. These different styles of packing are used both on piston and valve rods.

Fig. 12. Types of Rubber Packing

It is to be borne in mind that all which has been said with reference to the piston rod is equally applicable to valve stem packing. The general construction of the valve stem glands, vibrating cups, etc., is identical with those of the piston rod. The same materials are used for the packing medium and the same watchful care is required in order to obtain satisfactory results. Packing is an important subject and one which should be carefully looked after. It can not be said that any one particular kind or style of packing is the proper one to use in every case, for a packing which may give very satisfactory results under one set of conditions may utterly fail under another. For instance, a packing suitable for low steam pressures is not efficient where high steam pressures are used, and a packing that may give satisfaction with high pressures may not in any measure meet the requirements imposed upon it by the use of superheated steam. Each particular installation is, therefore, a different problem and must be solved in a different manner.

Valves. In Fig. 8, the valve *11* is shown in position. It will be noted that the valve rests upon the valve seat *14* and works between

the valve seats and the pressure plate *12*. The valve *11* is usually made of cast iron and may be of many different shapes, as will be seen in the study of the various types of engines. There are, however, two general types of valves—one, a plain slide or D-valve; and the other some form of piston valve. While there are many modifications and combinations of these two types, yet they are akin to the two types named. The valve in Fig. 8 is of the slide valve type. It is what is known as a double ported valve, that is, steam is admitted to the cylinder by two edges of the valve by reason of the fact that there is an opening through the valve. The pressure plate *12* is used for the purpose of reducing the area of the valve exposed to live steam pressure, it being noted that the portion under the hollow space *24* is not in contact with live steam. This reduc-

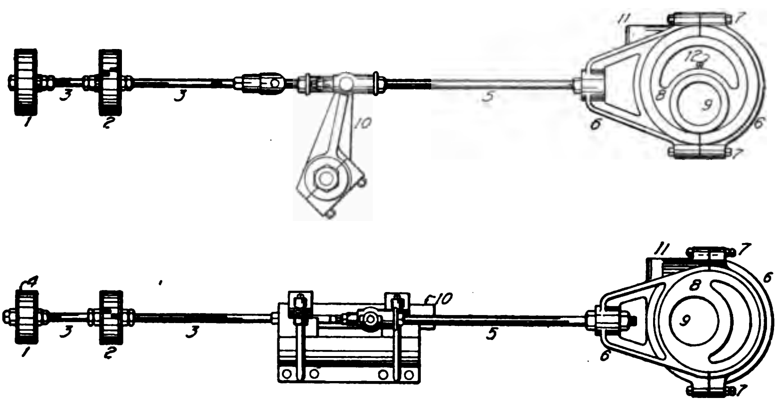


Fig. 13. Eccentric Mechanism Showing Rocker and Ram Methods of Connecting Eccentric Rod with Valve Rod

tion of the exposed area is made in order to reduce the amount of effort required to pull the valve back and forth. When the surface of a valve of medium size is considered and an average steam pressure per square inch of 180 pounds is being exerted upon it, some conception can then be had of the amount of friction that must be overcome every time the valve is moved across its seat. To eliminate a portion of this negative work is the primary object of the pressure plate. Pressure plates are of various shapes and designs depending of course upon the type of the engine and the valve used.

One of the advantages the piston valve has over the slide valve is that it is almost perfectly balanced by reason of the fact that steam

surrounds it on all sides, hence there is no excessive pressure on any part of the valve. It is comparatively light and therefore easily driven and lubricated. The general form of construction of piston valves is illustrated in plan in Figs. 21 and 22.

Eccentric. The valve is driven by its connection to the shaft by means of the valve stem, eccentric rod, and the eccentric. The relation of these parts is well illustrated in Fig. 13. The valve shown is an ordinary piston valve with flexible snap packing rings *4* similar to those previously described for the piston packing rings. In fact, the piston valve, as the name implies, behaves very much like the steam engine piston. The two piston ends *1* and *2* are held together by the valve rod *3*. The valve rod has nuts so placed that the pistons are held the proper distance apart. The valve rods, or stems as they are often called, extend beyond the valve box some distance and connect with the eccentric rod *5*. The manner of making the connection between the valve rod and the eccentric rod varies widely, this connection being governed largely by the type of engine and the exigencies of the case. Fig. 13 shows two methods of making this connection, one being accomplished by making use of a rocker arm and the other by using a ram. The way in which the rocker arm *10* is used, is obvious from the figure. The ram *10* is a square block, working in a bearing and so constructed that the valve and eccentric rod can be attached to it. When the ram is used, the motion is transmitted to the valve in a straight line, hence there is less strain upon the connecting parts than if a rocker arm was employed.

The eccentric rod *5*, in both cases, is attached at one end to the eccentric strap *6* and at the other end to the ram or rocker arm. Nuts suitably arranged make the rod secure and at the same time provide a means for lengthening or shortening the rod as needs demand. The valve and eccentric rod are usually made of mild steel turned true and polished.

The eccentric strap *6* is made of gray cast iron, lined with good babbitt metal for a wearing surface upon the eccentric. The strap is held on the eccentric by means of the bolts *7*. By removing liners or shims from between the two sections of the strap, adjustments for wear can be made. There are several patented straps on the market that possess particular features, but the essential elements

of all eccentric straps are about the same. Provision is made for lubrication by having an oil cup *11* cast with the strap.

The eccentric *8* is mounted on the main shaft *9* and is held secure in the position desired by means of the set screw *12*. Eccentrics for large engines are held by means of one or more set screws and a key. For a discussion of the function of the eccentric, the student is referred to the instruction book on "Valve Gears."

Steam Chest. The box *15*, Fig. 8, containing the valve and its parts, is known as the steam chest. The steam chest cover *16* is held in place by studs which pass through the flanges *17* into the box. The steam chest is connected to the steam supply by suitable pipe connections, steam being turned on or off as desired by means of the throttle valve. When the throttle valve is opened, steam passes into the chest through the valve, into the cylinder, where it is

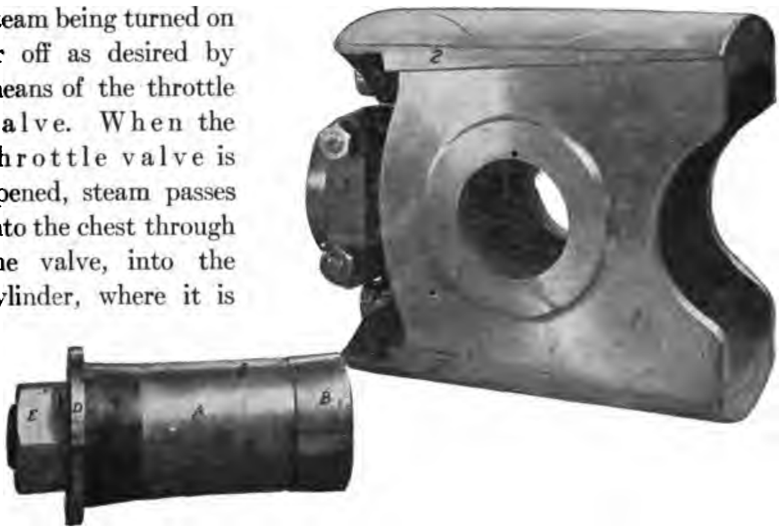


Fig. 14. Typical Crosshead and Pin for Large Size Engine

expanded and then ejected through the exhaust opening. The energy of the steam is transmitted through the piston and piston rod to the crosshead *17*, Fig. 6, thence to the connecting rod *23*, crank pin *33*, to the main shaft. In order that these parts may properly perform the function of transmitting this energy, a correct design is highly essential; therefore, a discussion of their construction is deemed necessary.

Crosshead and Connecting Rod. The crosshead is usually made of steel which forms a connecting link between the piston rod and

the connecting rod. It is made in various shapes and patterns. One type is illustrated in Fig. 6. In engines of larger size, the prevailing form of the crosshead used is similar to that illustrated in Fig. 14. This crosshead consists of a steel casting *1* and two wedges, or shoes, *2*, which fit over a projection on the outside surface of *1*. These wedges are either cast or forged and serve as a retainer for a layer of babbitt metal on the outside. It will be noted that there are oil grooves cut on the surface of *2* in order to facilitate the oiling of the crosshead guides. These wedges are provided with a nut and bolt *4*, whereby adjustment for wear can be made as necessary. Usually there is a slight amount of clearance between the crosshead and the guides, but it should not be in any case excessive. The piston rod is fitted into the end *3*, as already described. The connecting rod is attached to the crosshead pin *6*, which fits into the hole *5* and is held in place by a nut. The crosshead pin *6* is made of a good grade of steel and has a portion *B* which fits into the

Fig. 15. Solid Forged Steel Connecting Rod

back side of the crosshead, as viewed, the other portion *C* fitting into the outside part. When the pin is in place, the collar *D* is adjusted and the nut *E* tightly drawn. The straight portion *A* goes between the sides of the crosshead, and upon it the connecting rod brasses bear.

There are two general types of connecting rods in use, usually classified as marine and locomotive. Connecting rods of the marine type are as a rule used on engines of comparatively short stroke, while those of the locomotive type are employed on engines having a long stroke.

These rods are forged from open-hearth steel, with solid forged ends for the crosshead end, and a square end for the crank end in case of the marine type; and a solid forged or forked end for the crank end in case of the locomotive type.

The connecting rod, Fig. 15, is a solid forged steel rod having the ends machined out to receive the brasses. The crank end *1* is

fitted with a brass or bronze box lined with a good quality of babbitt metal. The crosshead end *2* is usually, but not always, fitted in a similar manner to that of the crank end. Adjustment for wear is made by means of wedges at each end, as shown at *3*. These rods are usually of rectangular cross section, although round shapes sometimes are used, especially on small engines.

The marine type of connecting rod is illustrated in Fig. 16. The body of the rod is forged similar to the locomotive type, as is also the small, or crosshead, end, but the distinguishing difference is in the way in which the large, or crank, end is formed. The end of the rod is enlarged and finished square, and the box containing the crank bearing which is lined with a good wearing material, is fastened to the rod proper by means of the bolts. Adjustment for wear is made by tightening up the nuts on the bolts.

It will be seen in Fig. 6 that the connecting rod is the connecting link between the crosshead and the crank *33*. The length of the

Fig. 16. Marine Type of Connecting Rod

connecting rod bears a definite relation to the length of the crank radius. The ratio of the length of the connecting rod to that of the crank radius varies in practice from four to eight. Occasionally conditions demand a greater ratio than eight, but it is seldom less than four.

Fig. 17 illustrates the connection of the piston, crosshead, connecting rod, and crank shaft. The function and construction of the piston, crosshead, and connecting rod have been previously discussed. However, the figure is valuable in that it shows quite clearly the relation of the various parts to each other. The crank shaft used on center-crank engines is frequently a solid steel forging, which includes the crank pin *2*.

Miscellaneous Parts. In order to compensate for the weight of the connecting rod and brasses it is necessary to put counterweights on the shaft as shown at *4*, Fig. 17. These counterweights are

usually heavy castings, machined to slip over projections on the crank shaft, and securely fastened thereto by bolts or set screws. The portion of the shaft marked *1*, Fig. 17, fits into the bearings provided for the main shaft or crank shaft, the length of this bearing portion being the distance between the counterweights and the collars *5*. It will be noted that on one end of the shaft is located a disk *3*. Sometimes this disk is forged as a part of the shaft and at other times it is made separate and forced on by hydraulic pressure. The purpose of this disk is usually intended to provide a ready

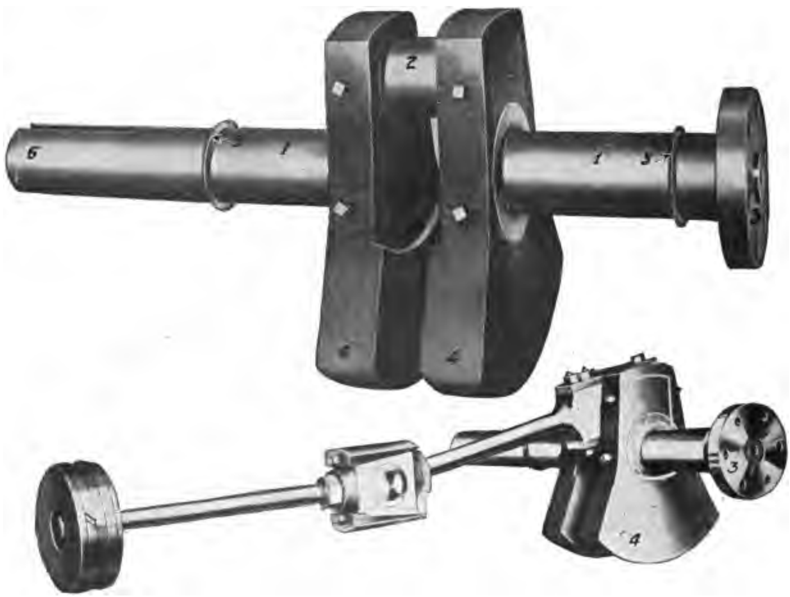


Fig. 17. Connection of Piston, Crosshead, Connecting Rod, and Crank Shaft

means of attaching the shaft of an electric generator when a direct connected plant is feasible or desired. It may be said here that a direct connected plant offers many advantages over a belt-driven system. It simplifies the plant, reduces friction, gives greater reliability, and makes possible more power in a given space.

The projection *6* on the other end of the shaft is the axis upon which the flywheel is forced and held secure by means of a key. The crank pin *2* should be of such ample proportions as to be safe against breakage and the heating of the pin or brasses placed upon it.

All engines are not of the center-crank type, but many have a side crank, the crank being a disk or a crank arm fastened on the end of the main shaft very much in the same manner as the disk *3*, Fig. 17. In this kind of construction the crank pin is usually a piece separate from the crank arm or crank disk, and is connected to it by being forced on and then riveted over, or by nuts put on and cottered. In either the side crank or center crank construction, the distance from the center of the axle to the center of the crank pin is equal to one-half the stroke of the engine, as for instance, an 18×24 engine has a crank arm of 12 inches in length, which is just one-half of the length of the stroke. In speaking of the size of the engine it is customary to mention the diameter of the cylinder first, that is, in speaking of an 18×24 engine is meant a cylinder 18 inches in diameter and a stroke 24 inches.

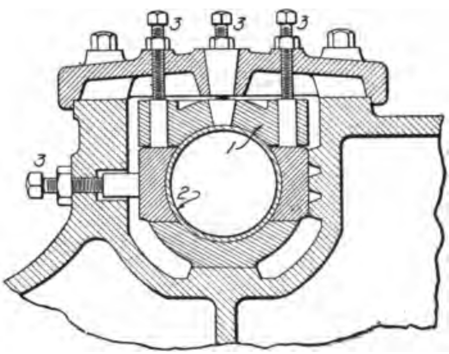


Fig. 18. Section of Babbitt Lined, Quarter-Boxed Main Bearing

The main bearing *4*, Fig. 7, should be designed with great care, having liberal proportions and lined with anti-friction metal, hammered in place and accurately bored and scraped to fit the shaft. On small engines the lower half of the main bearings are usually made of a part of the frame, the upper half being a removable cap. Between the upper and the lower portion of the bearing, metal liners are placed, which afford ready means for making any necessary adjustments.

Large engines have a babbitt lined, quarter-boxed main bearing of ample size, Fig. 18. To provide for both vertical and lateral adjustments it consists of four parts carefully machined on all sides and scraped to fit accurately. This bearing is so constructed that the bottom piece can be removed by slightly raising the shaft. The other three parts are removed after taking off the cap. By use of the adjusting screws *3*, the side *2* and the top *1* may be properly adjusted by the sense of feeling when the engine is in motion.

There are many other types of main bearings besides those mentioned, but they differ only from those already described in some of the minor details. The value of these details varies through wide limits, each builder contending for his own particular design.

A side-crank engine needs but one heavy bearing, such as that shown in Fig. 18, as the flywheel end of the shaft, being subjected to forces acting in but one direction only, requires a much smaller bearing. This outer bearing, Fig. 19, is called an out-board bearing and is smaller and simpler in construction than the main bearing. It is supported by a special casting, which has a hollow recess into which lubricating oil is poured. The shaft carries one or more small chains or rings which fit loosely on the shaft and dip into the

Fig. 19. Out-Board Bearing of Simpler Construction than Main Bearing

oil. Thus it is seen that oil is constantly brought in contact with the bearing of the shaft. This same scheme of lubrication is also used for the main bearing. As the different types of engines are considered, the several types of bearings will be noted and discussed.

The belt wheels *34*, Fig. 5, serve a two-fold purpose—one as a governing device, the value of which will be discussed later, and the other as a means of storing up energy while the piston is in mid-stroke, where the crank effort is greater than the resistance to be overcome. The belt wheels act as a flywheel and give up this energy at the ends of the stroke, thus enabling the engine to run over the dead centers. The design of the belt or flywheel is an important item in the proper proportioning of a steam engine. Its weight and dimensions must be very acurately determined. The belt wheel, or flywheel, whichever is employed, is made of cast iron

of various sizes, some being cast solid in one piece, others being cast in two or more sections. In any case the wheel is forced on the shaft and securely fastened thereto by means of a key and set screws.

TYPES AND CONSTRUCTION

Classification. Thus far an effort has been made to give the student some idea of the development of the steam engine and enable him to become familiar with the various parts and their functions. The natural sequence to the above study is to make a detailed study of the several types in use. No hard and fast rule can be given for classifying steam engines as they overlap in so many instances. That is to say, a simple engine may be either condensing or non-condensing; it may be high speed or low speed, etc. According to well-known authorities, various piston engines may be grouped under the following classes:

- I. Number of cylinders
  - Single cylinder
  - Multiple cylinder
- II. Construction of cylinders
  - Fixed cylinder
    - Vertical
    - Horizontal
    - Inclined
  - Movable cylinder
    - Oscillating
    - Rotary
- III. Action of steam
  - Single acting
  - Double acting
- IV. Transmission of steam power
  - Direct acting
  - Indirect acting
    - with balance lever or beam
    - without balance lever or beam

Professor R. H. Thurston in his book entitled "A Manual of the Steam Engine" classifies steam engines according to their purpose and use, as follows:

- I. Stationary mill engines
  - Moderate speed
  - High speed
- II. Agriculture engines
- III. Portable and semi-portable engines
- IV. Road locomotives
- V. Railway locomotives
- VI. Pumping engines
  - Crank and flywheel
  - Direct acting
- VII. Marine engines
  - Paddle engines
  - Screw engines
- VIII. Special types

The same authority further classifies engines according to their structure, as follows:

- I. Expansion
  - Simple
  - Compound
- II. Position of cylinder
  - Direct acting
  - Beam
  - Vertical
  - Inverted
  - Horizontal
  - Inclined
- III. Steam
  - Condensing
  - Non-condensing
- IV. Pressure
  - High pressure
  - Low pressure
- V. Piston action
  - Reciprocating
  - Vibrating
- VI. Steam turbines
- VII. Rotary
- VIII. Connection
  - Direct connected
  - Geared
- IX. Condensation
  - Jet condensing
  - Surface condensing

They are frequently designated by the name of the inventor, designer, or constructor, as the Watt, the Corliss, or the Porter engine.

From the last two groupings it is evident there is no sharp line of demarcation, for in many instances engines of one class have essential parts similar to those of another type. In this work the classification outlined in the last group will be taken as a basis for study.

Simple Engines. The simplest type of engine is the single expansion. It has one cylinder and admits steam for a part of the stroke, expands it during the remainder, and exhausts either into the atmosphere or into a condenser. Simple engines, Figs. 5 and 6, are now used only for comparatively small powers, say 200 h.p. or less, and although more extravagant in the use of fuel than the others, may still be the most economical financially, if low first cost is an important item; if they are not run continuously; or if the load fluctuates widely.

Compound Engines. Compound engines have two cylinders known as the high pressure and low pressure, Figs. 20 and 21. It will be noted that two different types of compounds are represented, the one in Fig. 20 being known as a cross-compound, the two cylinders

being parallel, and the one in Fig. 21, a tandem-compound engine, the cylinders being in line with each other.

Steam enters the smaller or high pressure cylinder, and then expands until release, when it is exhausted into the larger cylinder, where it expands further. The cylinders should be so proportioned that approximately the same amount of work can be done in each, which may be accomplished by making the high pressure cylinder enough smaller than the low so that when the steam leaves the high at a lower pressure than when it entered it, the increased volume of the steam may be taken care of and at the same time the increased area of the low pressure piston may compensate for the drop in steam pressure.

Fig. 20. Typical Cross-Compound Engine

Besides being economical, the cross-compound has a distinct mechanical advantage. The two cranks may be set at right angles so that when one is on dead center, the other is at a position of nearly its greatest effort. This makes a dead center impossible, and gives a more uniform turning moment. Then the individual parts may be made lighter and are thus more easily handled.

When the cranks of the cross-compound engine are at 90 degrees with each other the low pressure piston is not ready to receive the steam when the high pressure exhausts; therefore, there must be a receiver to hold the steam until admission occurs in the low. Such engines are called cross-compound, because steam crosses over from

one side to the other. Sometimes instead of having the cranks at 90 degrees, they are placed together or opposite. Then the strokes begin and end together, and the high can exhaust directly into the low without a receiver.

A tandem-compound engine, Fig. 21, has both pistons on one rod, the high pressure piston rod forming the low pressure tail rod. Such engines are less expensive because there is but one set of reciprocating parts instead of two, but like simple engines they have the disadvantage of dead points.

Triple Expansion Engines. Triple expansion engines expand the steam in three stages instead of two. There are usually three

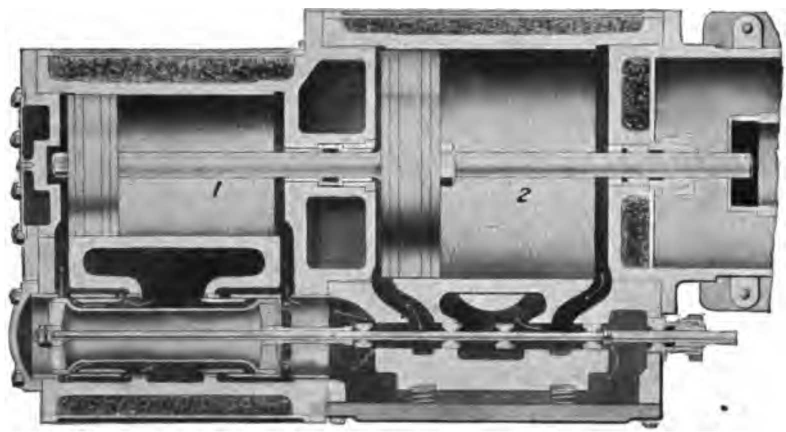


Fig. 21. Section of Cylinder and Valves of a Tandem-Compound Engine

cylinders, viz, the high, the intermediate, and the low, arranged with cranks 120 degrees apart. This gives a more uniform turning moment than a compound. Sometimes there are four cylinders on the triple expansion engine, viz, one high, one intermediate, and two low. This arrangement gives better balance and is often used in marine work.

For triple engines there must be a receiver between each two cylinders. Fig. 22 shows the essential features of a triple expansion engine.

Quadruple Engines. Quadruple engines expand their steam in four stages instead of three. Multiple expansion engines are nearly always condensing.

Cylinder Ratios. There are several considerations to be remembered when proportioning the cylinders of the multiple expansion engines. The ratio of the cylinders should be such that each develops nearly the same power, and the drop in pressure between the cylinders and receivers should be as small as possible.

There are many formulas in use, some simple, others more complex involving mathematical calculation. A common rule for compound engines is to make the ratio of the cylinders equal to the square root of the total ratio of expansion. Thus, if the steam has an expan-

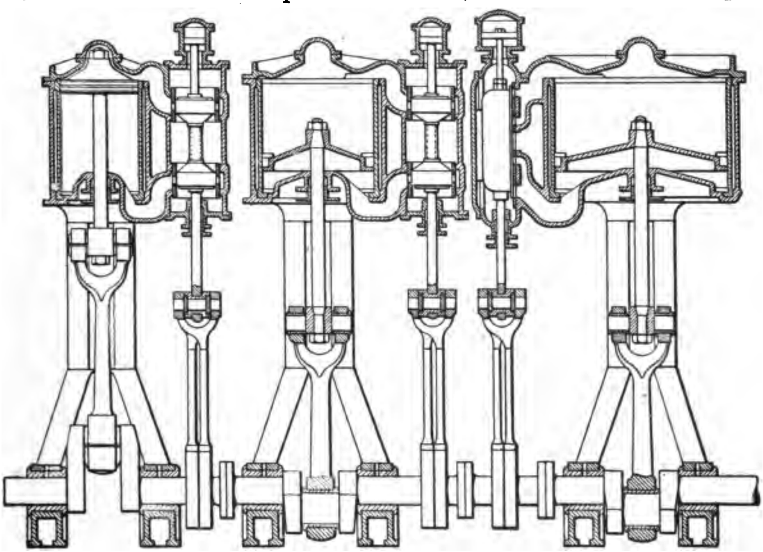


Fig. 22. Section of Essential Features of Triple Expansion Engine

sion ratio of 9, the ratio of the cylinder volumes will be $\sqrt{9}$, or 3; that is, the low pressure cylinder will have a volume three times as great as the high pressure cylinder. If the cylinder ratio is 3 and the length of the stroke is the same for both, the diameter of the low pressure cylinder will be 1.75 times that of the high pressure cylinder.

Another rule is to make the cylinder ratio equal to the total ratio of expansion multiplied by the fractional part of the stroke completed when cut-off occurs in the high pressure cylinder.

Suppose the ratio of expansion is 9, as above, and that cut-off occurs at one-third of the stroke in the high pressure cylinder, the ratio of cylinder volumes will be $9\times\frac{1}{3}$, or 3. If cut-off occurs at one-half of the stroke, the ratio will be $9\times\frac{1}{2}$, or 4.5.

For triple expansion engines the low pressure cylinder is made large enough to develop the full power if steam at boiler pressure is used.

The intermediate cylinder is made approximately a mean between the high and low. The area of the intermediate piston is found by dividing the area of the low by one and one-tenth times the square root of the ratio of the low to the high.

The above may be written thus:

$$\text{Area of high pressure cylinder} = \frac{\text{Area of low pressure cylinder}}{\text{Cut-off of high pressure} \times \text{ratio of exp.}}$$

$$\text{Area of inter. cyl.} = \frac{\text{Area of low pressure cylinder}}{1.1 \times \sqrt{\text{ratio of low to high}}}$$

In general, for triple expansion the ratios of the volume of the three cylinders are about as follows:

$$V_1 : V_2 : V_3 :: 1 : 2.25 \text{ to } 2.75 : 5 \text{ to } 8$$

For quadruple expansion engines, the ratios are as follows:

$$V_1 : V_2 : V_3 : V_4 :: 1 : 2 \text{ to } 2.33 : 4 \text{ to } 5 : 7 \text{ to } 12$$

It is self-evident that the compound engines illustrated are of the multiple cylinder class. They also have fixed cylinders and are double and direct acting. That is, steam acts on both sides of the piston, and the power is delivered directly from the piston to the shaft or flywheel without the intervention of a walking beam or some other transmitting medium. The engines illustrated in Figs. 20 and 21, are horizontal, whereas the one shown in Fig. 22 is vertical. A horizontal engine is, therefore, an engine whose cylinder is parallel to the ground, and a vertical engine is one which has its cylinder or cylinders perpendicular to the ground. These engines may also be operated either condensing or non-condensing.

From the foregoing it must be obvious that it is not possible to classify an engine within narrow limits, so it appears to be more logical to classify them according to the service for which they are to be used, as in the second grouping.

Selection of Type. In the selection and design of an engine there are a great many factors to be considered. The engine must be as light as possible, and yet must be strong enough to do the work

likely to be imposed upon it. The bearings should be large and ample in number. Lubrication must be given especial attention if high speeds are to be used. Lightness of design tends towards small first cost, which is important, but durability and efficiency should not be entirely sacrificed for low first cost. In the course of time the more expensive engine may prove to be the cheaper as maintenance and repairs may amount to considerable on a poorly designed and built engine. For some classes of service, however, the cheap engine is the one best adapted. For instance, in saw mills, cotton mills, and for similar class of service, a low first cost simple engine is the one best suited for the work, because the labor employed to operate it is often inexperienced and ignorant. In such cases the protection and care that can be given the engine is poor, hence the lower the value of the property exposed; the less will be the loss resulting from the depreciation. On the other hand, if one is selecting an engine for a lighting plant in a city, he would more than likely select one of the most improved types of high speed, condensing machines. In the latter case the first cost would be considerably more than the one selected for the saw mill, but the increased efficiency of operation, the slight depreciation, and the reduction in maintenance would more than compensate for this.

From the foregoing it is evident that there are many factors to be taken into consideration when selecting a steam engine for any given service. In the further study of the several different types the class of service for which each is best suited will be indicated in so far as it is possible to do so. There are, however, some general features every engine should possess independent of its class. It should be simple in construction, having compactness combined with great strength and durability. It should be well balanced and free from severe vibration. Accessibility of parts is also an important consideration.

STATIONARY ENGINES

Simple Side-Crank Type. Stationary engines for ordinary mill service, such as machine shops, small power plants, and various manufacturing concerns, are generally simple engines operating at moderate speed, having either plain slide valves or piston valves. There are, however, some cases where compound engines of moder-

ate size have been installed in similar plants in more recent years. The demand for electric generators has also largely affected the design of steam engines for small electric power plants. In speaking of small plants, in this connection, it may be taken as meaning from 25 to 500 horsepower.

A simple slide valve engine of the side-crank type, which has been largely used in plants where a cheap, efficient engine was the requirement, is illustrated in Fig. 23. This engine has one slide valve, an automatic or shaft governor, and a heavy flywheel which is used as a belt pulley. It is built in sizes varying from 9 inches × 14 inches to 22 inches × 28 inches, and develops a horsepower of

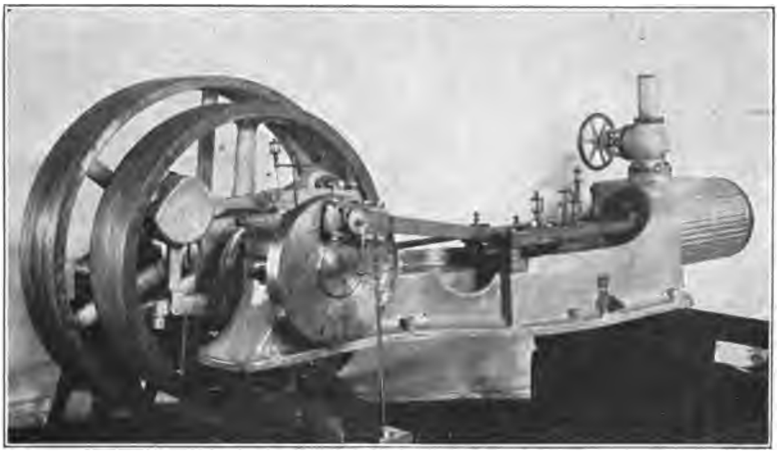


Fig. 23. Simple Slide Valve Engine of Side-Crank Type

about 45 to 300 according to size of cylinders, steam pressure used, and the speed at which the engine is operated.

Lubrication of the cylinders is secured by the use of a sight feed lubrication attached to the steam pipe. The main and crosshead bearings are lubricated by oil cups.

This type of engine has been extensively used in cotton gins and saw mills and in small machine shops throughout the country. There are, however, several grades on the market, and it may be purchased for a comparatively low figure where the work to be done does not demand a machine of high grade. This engine has a concrete foundation, is well proportioned, and makes a neat appearance.

Simple Vertical Type. A simple vertical high speed engine that is particularly well adapted for isolated lighting plants in factories, stores, mines, and aboard ships, is illustrated in cross section in Fig. 24. It requires little attention, occupies small floor space, and is not extravagant in the use of steam. The engine is neatly and well designed. It has a large base, which insures stability and rigidity. All of the working parts are enclosed, but readily accessible for inspection and repairs. The frame, cylinders, valves, pistons, etc., are carefully made and adjusted, and the same general types of these various parts conform to the general practice of high speed engines. It will be noted from the illustration that it has a center-crank, automatic governor, and a piston valve. The lubrication of the moving parts is accomplished by means of a geared pump located in the interior of the base of the frame. This pump forces the oil through pipes and grooves to the various bearings. This type of engine is furnished by the makers in sizes from $3\frac{1}{4}$ inches × 3 inches up to 9 inches × 7 inches for general service. Much larger vertical engines may be obtained, but are made as a special order. The engine illustrated is so designed to operate at speeds from 250 to 500 revolutions per minute, depending on the size, and uses steam pressure from 60 to 150 pounds. Its commercial rating is from $1\frac{1}{2}$ to 60 horsepower, according to the size of cylinders, steam pressure, and speed of operation.

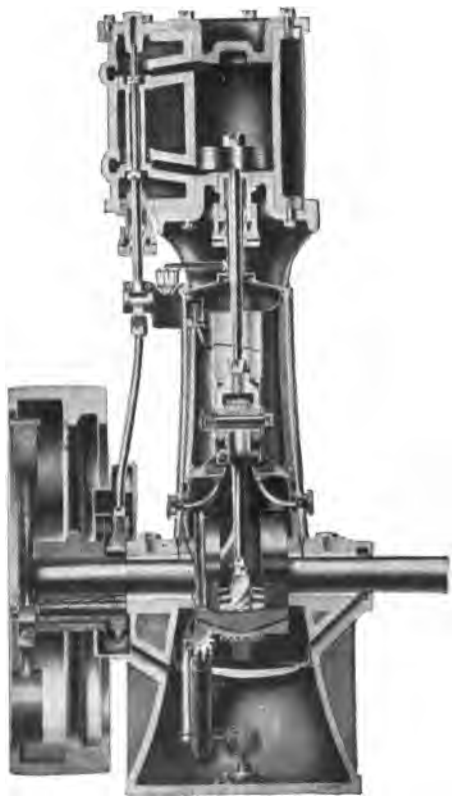


Fig. 24. Vertical High Speed Engine

Advantages of Vertical over Horizontal Type. While the discussion given above has had to do with a vertical engine of rather small dimensions and power, yet it must be borne in mind that vertical engines in very large units are built and successfully operated. This leads to a discussion of the relative merits of horizontal and vertical engines. At the present time the most common type of engine is the horizontal direct-acting, that is, an engine whose cylinder is horizontal and whose piston acts on the crank through a piston rod and a connecting rod. In small engines the whole is often on one bed plate. Such engines are said to be self-contained. The cylinder is either bolted to the back of the bed plate or rests directly on it.

In marine work vertical engines are used in almost every case, on account of the *saving of floor space,* which is so important in a vessel. This saving of space is also a very important factor in many other cases, such as in crowded engine rooms in cities where land is expensive.

A second advantage of the vertical over the horizontal engine is the *reduction of the cylinder friction and unequal wear* in the cylinder of the latter. In the horizontal engine the piston is generally supported by resting on the cylinder, which is gradually worn until it is no longer round, causing leakage of steam from one side to the other. This is entirely avoided in the vertical engine.

Still another advantage of the vertical engine is the *greater ease of balancing the moving parts* so that there shall be no jarring or shaking. It is impossible to perfectly balance a steam engine of one or two cylinders. If it is balanced so there is no tendency to shake sidewise it will shake endwise; and if it is balanced endwise it will shake sidewise. The jarring is due to the back and forth motion of the reciprocating parts and the centrifugal force of the crank and the connecting rod. The crank can be readily balanced by making it extend as far on one side of the shaft as it does on the other, but the piston and the connecting rod are more difficult to balance. The effect of jarring can be greatly reduced, if the crank be balanced and the endwise throw made to come in line with the foundation, which should be heavy enough to absorb the vibration transmitted. In a horizontal engine this endwise throw not being in line with the foundation will cause vibration in the engine itself.

In machines that can be anchored down to a massive foundation, a state of defective balance only results in straining the parts and

causing needless wear and friction at the crank-shaft bearings and elsewhere, and in communicating some tremor to the ground. The problem of balancing is much more of consequence in locomotive and marine engines.

To sum up the general advantages of the vertical engines: they

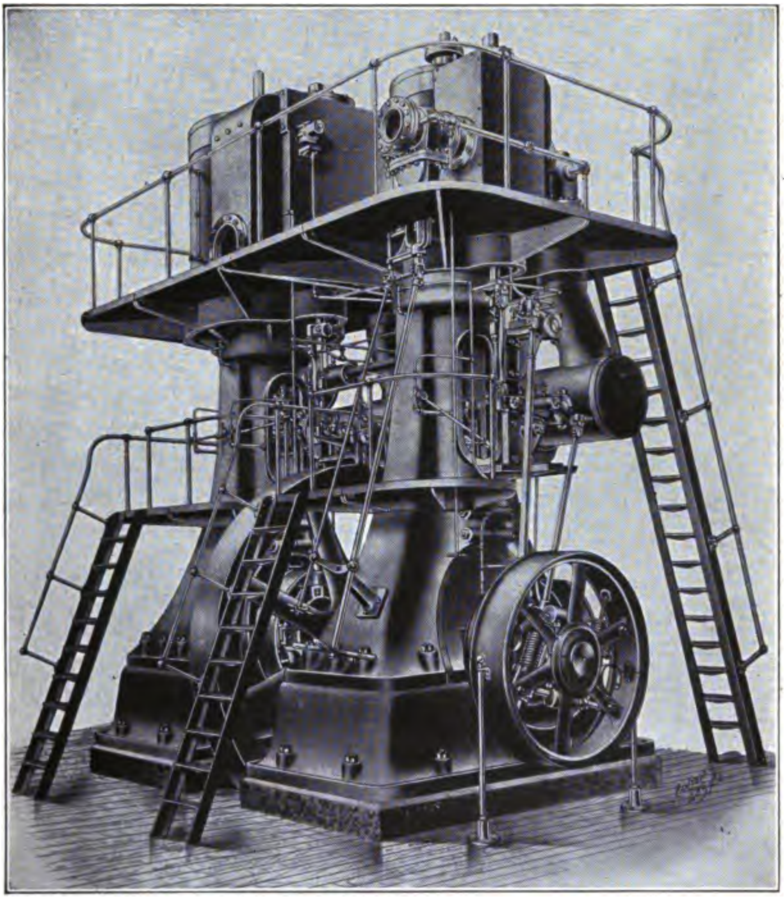


Fig. 25. Buckeye Vertical Cross-Compound Engine

have less cylinder wear, they take up less floor space, and they can be better balanced. In addition to these there are certain advantages which vertical engines have for certain kinds of work.

Disadvantages of Vertical Type. The *pressure on the crank pin* is greater during the down stroke than during the up stroke, because

during the down stroke the weight of the reciprocating parts is added to the steam pressure, and during the up stroke this weight is subtracted.

Another difficulty is that in large engines the various parts are on such different levels that they require considerable climbing. This requires *more attendants* and is sometimes the cause for neglect of the engine. The *foundations* for vertical engines need to be deeper than those for horizontal engines, yet they do not need to be as broad.

Buckeye Vertical Cross-Compound Type. The development of electrical machinery and the increased demand for power in con-

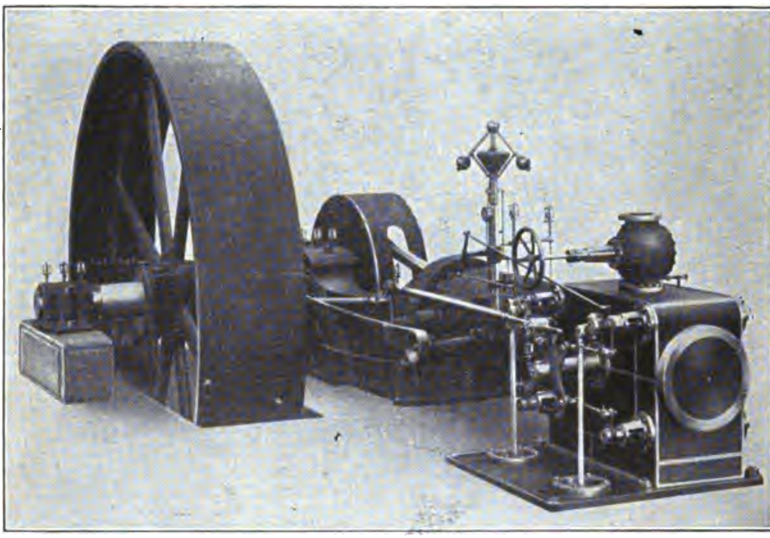


Fig. 26. Simple Corliss Engine, Showing Valve Mechanism

gested city locations, where land is very expensive and buildings costly because of their great height, has been the primary cause of the development of large vertical steam engines of various types. The engine, Fig. 25, represents a vertical cross-compound engine as built by the Buckeye Engine Company, which is especially well adapted to electric railway and power and lighting plants, when floor space is limited. The engine may be obtained either as a side or a center crank design. This engine and simple horizontal engines of the same make are typical representatives of economical, high speed engines. They are high priced, but the economy of operation and maintenance make them

very desirable. The vertical engine illustrated may be obtained in sizes developing from 75 to 3,000 horsepower. A discussion of the valve gear used on Buckeye engines is to be found in the instruction paper on "Valve Gears." It is a double valve, giving automatic cut-off as distinguished from throttling cut-off regulation.

Corliss Type. A general utility engine of the highest type both from the standpoint of design and economy of operation and maintenance is the Corliss engine. It is to be found in electric railway power stations; in large and small pumping stations; in blast furnaces and rolling mills; in textile and flour mills; in machine shops and office buildings; in technical schools and colleges; and in nearly all kinds of industrial plants in this country and abroad. The Corliss engine is built in various types, styles, and patterns of any designed capacity up to 10,000 horsepower.

Fig. 26 shows the valve connection and manner of operation of a simple Corliss engine. It will be noted that the engine is governed by a fly-ball governor which is driven by a belt connection to the main shaft. This governor is connected to the steam valves by reach rods. The speed is automatically governed by variation of the point of cut-off. This engine, as well as most engines of this type, has large well-proportioned frames, cylinders, etc. Good workmanship and material enter into its construction, hence it is known as a high-priced engine; but, on the other hand, it is perhaps the most economical in the use of steam.

Valve Mechanism. The distinguishing feature of the Corliss engine is its valve mechanism, a good view of which may be seen in Fig. 27. The gear has four valves, the two top ones being the admission or steam valves and the two lower ones the exhaust valves. There is a connecting rod *7*, Fig. 27, which is connected to the eccentric through a rocker arm, and another rod, as may be seen in Fig. 26. As the shaft revolves, the rod *7*, due to its connection to the eccentric, moves back and forth, and, by reason of its connection through the clamp *8* to the wrist plate *6*, the latter is made to oscillate. The wrist plate *6* is attached to the frame by a pivot projection. The rods *9* have a right and left screw adjustment on each end and transmit motion from the pins *14* on the wrist plate *6* to the steam and exhaust valve bell cranks *10* and *15*, respectively. These valves receive motion in such a manner as to open and close the ports rapidly.

The steam valve bell crank *10* is free to rotate on projections of the bonnet and carries at the end of the lever shown nearly horizontal the brass hook *3* which engages with the catch block. This catch block is rigidly attached to the valve lever *13*, which is keyed to the end of the valve stem, the latter transmitting motion to the valve. Attached to the valve lever *13* is the dashpot piston rod *4*. The hook is so made that it may be automatically tripped when the back part of the hook comes in contact with a cam which is operated by the

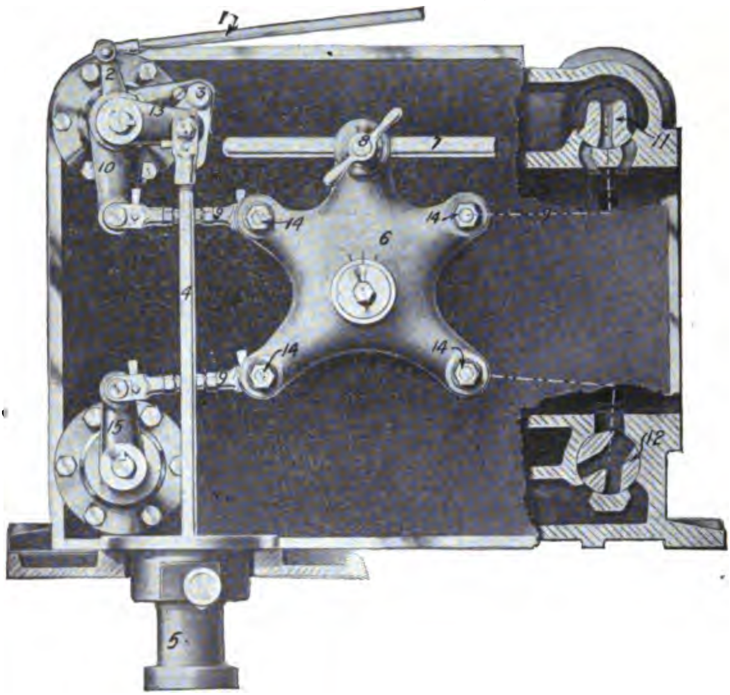


Fig. 27. Corliss Valve Mechanism in Detail

arm *2* connected to the governor by the reach rod *1*. The operation of the mechanism is such that the hook may be disengaged at any point of its travel by means of the cam coming in contact with the tripping leg of the hook *3* and causing it to rotate on the pin and move the steel catch out of engagement with the catch block.

The slowing down of the engine, in consequence of reduced steam pressure or an increased load, causes the catch to hold its contact longer and the steam to be admitted longer. In the event that the

speed be increased in consequence of increased steam pressure or diminished load, the hook would be tripped by the cam and the admission valve would be quickly closed by the vacuum dashpot *5*. It must be evident from the foregoing that the regulation obtained by this device must be very sensitive to any change of speed or load. The dashpot *5* closes the steam valve when the hook is tripped by the cam.

The cylinders have four cylindrical holes accurately bored at the four corners, as is shown at *11* and *12* in Fig. 27. Into these openings the valves are placed with their stems and proper packing devices. The seats of the valves are circular. The portion of the valve marked *2* and *1*, Fig. 28, is circular, whereas the remaining portion may have any shape, depending upon the requirements of the design. The valve stem *5-4-6* is also irregular in shape. The portion *4* fits into the slot *3* of the valve and round portions *5* and *6* serve as bearings and as means for attaching the driving mechanism.

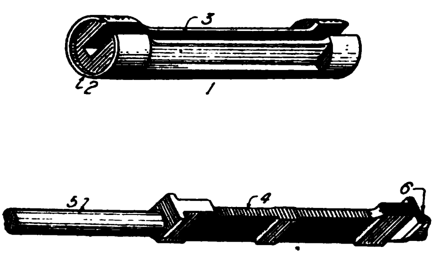


Fig. 28. Corliss Valve and Valve Stem

Advantages and Disadvantages of Corliss Type. Perhaps one of the chief disadvantages of the Corliss engine is the large amount of floor space required, a factor which often precludes its use. It possesses many advantages, however, chief among which may be mentioned the rapid and wide opening of the steam and exhaust ports; shortness and directness of ports, which results in small clearance; the adaptation of the steam valve to the functions of cut-off valves; and the location of the exhaust ports at the bottom side of the cylinder, thus draining the cylinders perfectly. Each of these various factors contribute to good engine performance, and their combination has resulted in making the Corliss engine one of the most economical engines manufactured. It will operate upon from sixteen to eighteen pounds of steam per indicated horsepower per hour.

Angle-Compound Type. As an outgrowth of the demand for an engine of high speed and one that will occupy a small space, but which, at the same time, will be economical in the use of steam, there has been developed the angle-compound engine shown in Fig. 29.

Balancing. In an ordinary high speed steam engine, the inertia of the reciprocating parts—namely, the crosshead, piston, and piston rod—and the crosshead end of the connecting rod, is considerable. If a steam engine is to be installed in office buildings, apartment houses, or in other houses where freedom from vibration is a prime requisite, it becomes almost a necessity for the engine to be perfectly balanced. On an ordinary reciprocating engine it is almost impossible to obtain perfect balancing for two reasons:

First, because of the angularity of the connecting rod, which causes the rate of acceleration of the reciprocating parts to be much faster at one end than the other, therefore, the counterweight which exactly

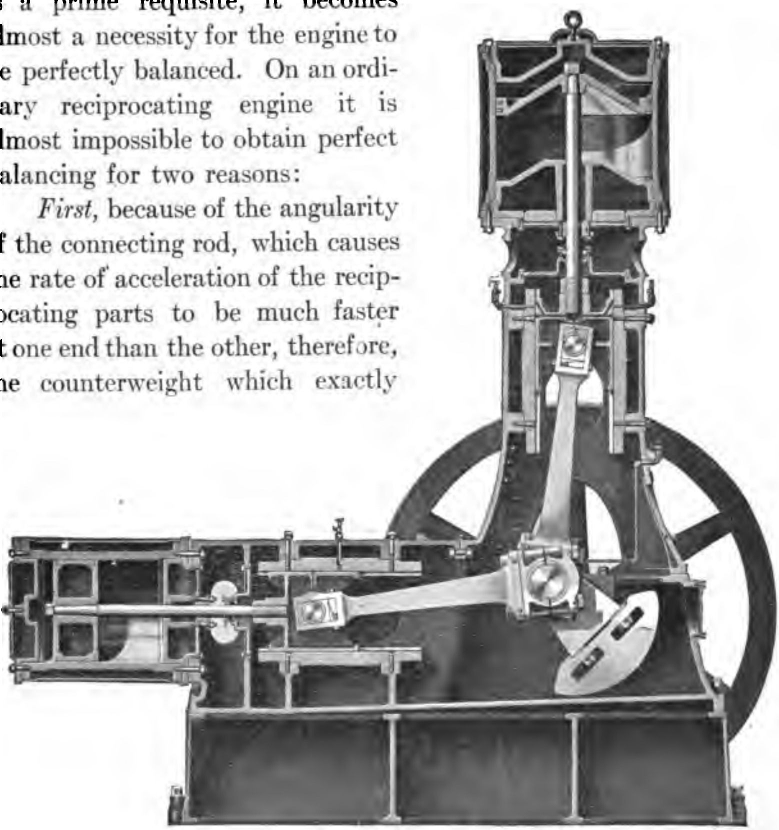


Fig. 29. Section of Angle-Compound Engine

balances the forces at one end would be either too light or too heavy at the other end.

Second, the counterweight at all positions in the revolution of the shaft exerts a radial force and when the counterweight is above or below the center of the shaft, there are no reciprocating parts developing a counteracting force, hence the centrifugal force of the

counterweight exerts a powerful unbalanced vertical force. (This has been observed a number of times in locomotive practice where the rails have been bent by the extremely heavy blows of the unbalanced forces.)

In tests at Purdue University on their locomotive testing plant, it was clearly demonstrated that the unbalanced vertical forces are so great at high speeds that the locomotive driver is at times lifted clear off the track. It is obvious from the foregoing that the question of balancing is a serious one, and one that should be carefully considered. A thorough study of the question would involve considerable time and space and the use of higher mathematics.

The several engine builders who put the angle-compound engine upon the market claim for it an elimination of the balancing difficulty. As will be seen from the illustration, the angle-compound consists in combining two engines in such a manner that one crank pin serves both. The high pressure and the low pressure cylinders are placed at 90 degrees from each other in the plane of rotation of the crank. The horizontal engine is arranged so that it is perfectly balanced along its horizontal axis, but is, of course, badly out of balance vertically. On the other hand, the vertical engine is perfectly balanced along its vertical axis, but is out of balance in a horizontal direction. The above statements are true only when we consider each engine separately. When the engines are placed together, the unbalanced effect on one tends to neutralize that of the other. Their relation is such that the same counterbalance serves for both engines. It is claimed for this arrangement that there are four points in the revolution where a perfect balance exists and the resultant effect is to give almost a perfect balance. Another point of interest with these engines is that there are no dead centers; hence by employing a by-pass connecting the two cylinders, the engine can be easily started from any position of the crank.

Summary of Advantages. This type of engine, therefore, possesses the advantage of good balancing; it occupies about one-half of the floor space of a simple engine of the same power; and the compounding reduces its steam consumption considerably below that of a plain slide valve engine.

General Survey of Stationary Types. The treatment of the subject of mill or stationary engines, in so far as the scope of this

work will permit, has been covered by the discussions given concerning the plain slide valve engine, the vertical engine of small units, the vertical engines for large installations, the compound and tandem engines which are being used more and more, and finally by a consideration of thé most economical engine of all, the Corliss, whether operated simple or compound. In addition to the types mentioned there are a large number of other makes, which have distinguishing features and which give good service, but yet the principles enumerated in the types already discussed fulfill all the requirements likely to be made upon stationary plants. Hence a discussion of other makes is not thought necessary.

FARM OR TRACTION ENGINE

The advancement of scientific and progressive farming has made the farm engine of more interest and importance than ever before; in fact, the demands of the active farmers in recent years have taxed the builders of such equipment to the limit of their output. The steam engine is used for a large variety of purposes upon the modern large farm, and appears most commonly in the form of the so-called traction engine. It is used for plowing, digging ditches, building of roadways, logging purposes, running threshers, and numerous other purposes. Various types of stationary engines of small power are also to be found in use on the farm, the small gas engines now having been perfected to such a degree that they are rapidly replacing the steam engine.

General Description. The traction engine is really more than simply an engine; in fact, it is a self-contained power plant. It consists of a simple or compound engine, a boiler for supplying the steam required by the engine, and the transmission mechanism, together with all the auxiliaries necessary for a complete power plant. A good type of a general utility traction engine is shown in Fig. 30. It consists of a boiler of the locomotive type, carried by four wheels, the two front ones serving as a means for guiding, and the two rear being the ones which receive the power and known as the driving wheels. In order to prevent the slipping of the rear wheels when doing heavy hauling, they are made with heavy projecting lugs or cleats which are forced into the ground by the weight of the machine. The engine, which is mounted on the side of the boiler, as may be

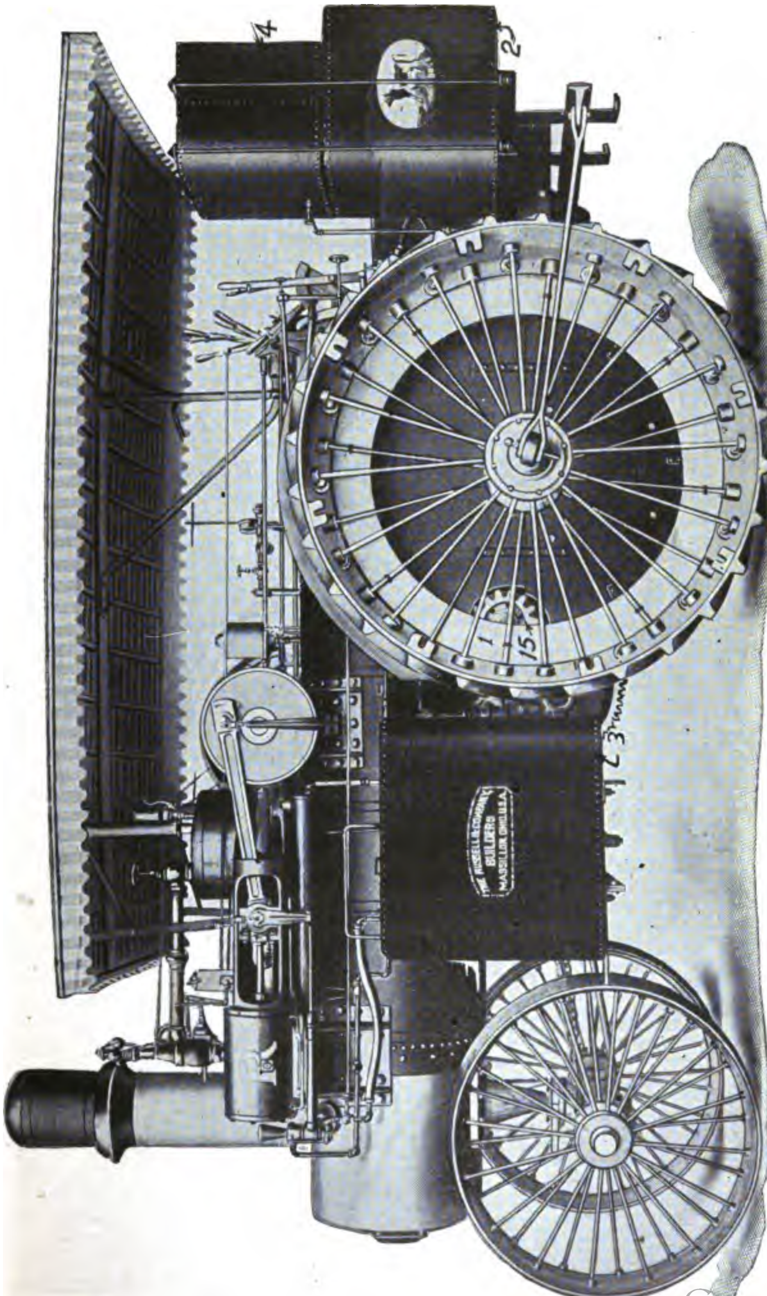


Fig. 30. Standard Type of Steam Traction Engine

seen in the illustration, is a plain slide valve engine of the side-crank type. The speed is regulated by an ordinary fly-ball, centrifugal governor. The construction of the various parts of the engine are similar to those previously described in this work. The same watchful care should be given to the lubrication, operation, and maintenance of this engine as to any other, when economy, durability, and reliability are desired. It should be noted that both cross-compound and tandemco-mpound engines are used as well as simple engines in this class of service, and that various types of valves find application.

In order to make clear the construction and operation of the traction engine, a view showing the rear wheels, platform, and side tanks removed is shown in Fig. 31. The means provided for guiding, reversing, and driving this engine is clearly illustrated. It is evident that the type of boiler used is similar to that of the locomotive boiler, having a narrow fire box. It has an extended front end *1* and stack *2* for carrying away the gases of combustion. The boiler is mounted upon the front wheels through the pivoted pedestal connection *3*. It is supported on the rear wheels, by having the rear axle extend beneath the fire box, or by having the supporting elements riveted to the side sheets as in Fig. 31.

Operation of Plant. *Reversing Mechanism.* The operation of the plant is about as follows: If the engineer desires to go forward the mechanism is placed in forward gear by means of the reversing lever *29*, the reversing being accomplished by means of a swinging eccentric, which can be thrown across the shaft at the discretion of the operator. (On some types of traction engines, a reversing link mechanism is used.)

Transmission. Having adjusted the reversing gear in accord with the desired direction, the throttle valve of the engine is opened by moving the lever *30*. The opening of the throttle valve starts the the engine shaft *12*, which carries the flywheel *11*. On the engine shaft behind the flywheel is keyed a small spur gear which is in mesh with the larger gear *13*, which in turn meshes with the gear *14*. As the engine shaft revolves, the small gear in the shaft revolves, which transmits its motion to *13* and on to the small gear *15*, which is keyed to the shaft driven by the wheel *14*. The gear wheel *15*, Figs. 30 and 31, is in mesh with an annular gear on the drive wheel *1*, Fig.

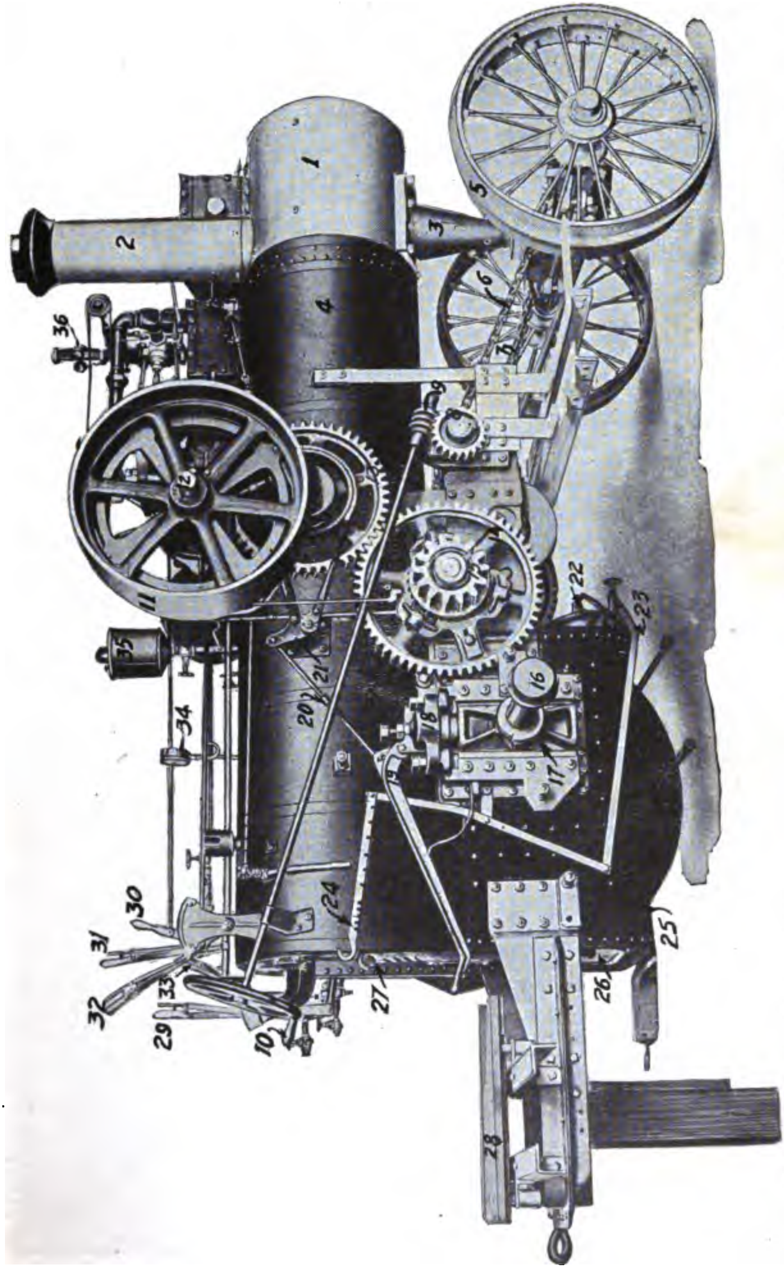


Fig. 31. Traction Engine with Rear Wheels, Platform and Side Tanks Removed to Show Steering and Driving Mechanism

30; hence, by reason of this connection, the large wheel is made to revolve. The shaft carrying gear wheel *15* extends beneath the boiler to the opposite side and drives a set of gear wheels which causes the other driving wheel to revolve with the one just considered.

Running Gear. The axle *16* of the wheel has a sliding head *17* attached to it. This head is free to move up and down in guides securely fastened to the fire box. This sliding head works against a spring, which is contained in the box *18.* This spring reduces the shocks to which the machine is subjected when on the road, hence, the engine is much easier to ride than it otherwise would be. In addition to the easy riding qualities, it also relieves the parts of the machine of stresses and strains due to sudden jolts, which would be detrimental to the durability of the machine as a whole.

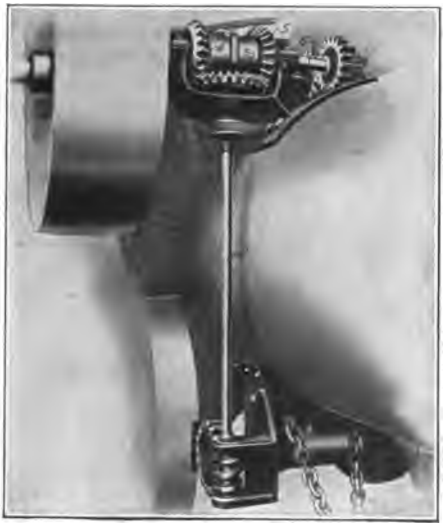


Fig. 32. Friction Gear Device for Steering Traction Engine

Steering Gear. The engine is guided by the hand wheel *10.* It will be noted that a chain is connected to the front axle on either side of the pivotal point. This chain wraps around a cam arrangement on the shaft which carries the small gear wheel *8.* The wheel *8* is in mesh with the worm *9,* which may be turned by the hand wheel *10.* If the driver when moving forward should wish to turn to the right, for instance, he would turn the hand wheel so the wheel *8* would be driven counter-clockwise, and in so doing the chain *6* would be shortened, the chain *7* lengthened, the wheel *5* would be cut in, and the machine would turn to the right. If it was desired to go in the opposite direction, the reverse operation would be carried out, that is, gear *8* would be revolved clockwise.

It is sometimes difficult to operate the steering gear by hand, especially in large traction engines and in places where a heavy load

is being driven over rough ground; hence, some engines are provided with a friction gear device. This attachment, Fig. 32, is exceedingly simple, and when it is used the engine furnishes power to guide itself. It consists of a shaft *2* extending from the worm gear *1* to a bracket on the side of the boiler in front of the main shaft. On top of this vertical shaft is a horizontal miter gear *3* arranged to engage alternately with two vertical gears, one at the right *5* and the other at the left *4*. These vertical gears are on a shaft run by a chain of small gearing *6* from the engine shaft. They are thrown in or out, at the pleasure of the operator, by means of a shifting yoke which is worked by a straight rod extending back to the right-hand side of the engineer. A lever at the end of this rod is within easy reach all the time. By moving it forward or backward, the engine is guided to the right or left, as desired. If the lever remains at the center, the engine guides straight. An extension rod is placed on the rear end connecting with a hand lever at the left side of the platform, so that the engine may be guided equally well from either side. To operate this steering lever requires no appreciable exertion on the part of the engineer.

Friction Clutch. A friction clutch is provided in the flywheel, which permits the engine to be operated without driving the machine forward on the road. With the engine running at full speed, the clutch can be gradually thrown into action, and the machine will start forward on the road without any sudden shocks. The clutch is operated by the lever *31*, Fig. 31. By disconnecting the engine from the flywheel, a high speed can be obtained, so that by throwing the clutch in gear quickly the engine is often able to pull the machine out of difficult places. Oftentimes it is desired to operate the engine independently of the traction wheels for the purpose of running the thresher, saw mill, electric generator, or for other purposes, hence some form of clutch is necessary.

Brake. A friction brake is operated by a system of levers and rods as *19*, *20*, and *21*, Fig. 31. The operator can apply the brake by pushing downward upon the foot piece on the lever *19*. The amount of air admitted to the fire box is controlled by the two dampers *22* and *26*, which may be manipulated by the levers *24* and *27*.

Water Tanks. In Fig. 30, large tanks *2*, *3*, and *4*, are shown. These tanks are water reservoirs from which the supply pumps take water and deliver it to the boiler. Opposite the tank *2* is a bin for

holding the fuel, which may be wood, coal, or straw, depending upon the location and character of work to be done. If the traction engine is used for threshing purposes, it would have a fire box arranged for burning straw; whereas if it was being used in a logging camp or a saw mill, the available fuel would be wood, hence the fire box would be constructed accordingly.

Boiler. Since it is necessary to have a high-grade, durable, and economical boiler in order to have an efficient and reliable machine, it is thought advisable to call especial attention to the type of boilers used in this connection and point out some of their good and bad features. It was mentioned in the description of the traction engine, Fig. 30, that a locomotive type of boiler with some modifications was used. Fig. 33 illustrates such a boiler. It is of the fire tube horizon-

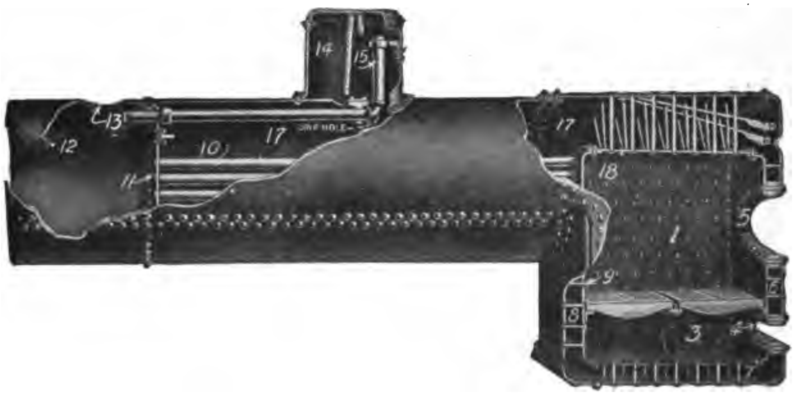


Fig. 33. Traction Boiler of the Locomotive Type

tal type. The fire box *1* is of a horizontal rectangular construction with open grate bars *2* and ash pit *3*, below. The fuel, either wood or coal, is fed through the fire door *5*, and the ash is removed through the door *4*. The products of combustion, such as smoke, hot gases, etc., pass through the tubes *10* into the front end *12*, from whence they are exhausted through the opening *13* and the smoke stack into the atmosphere. If straw is to be used as fuel, a brick arch is placed in the fire box which deflects the gases toward the fire door so that, after passing over the arch, they are drawn out through the tubes in the usual manner. It is also necessary to put in different grate bars where straw is used, as the bars must be closer together so the fuel will not drop through into the ash pan.

It will be noted that the fire box is surrounded by water legs *6*, *7*, and *8*, and the water and steam space *17*. Water is also circulating around the tubes and is several inches deep above the crown sheet *18*. As combustion takes place in the fire box and the hot gases pass through the tubes, the plates of the fire box and the tubes become heated. As a consequence, the water in contact with these hot surfaces becomes heated also, and steam is formed which rises to the top of the boiler, entering the steam dome *14* from whence it is taken by the pipe *15* through a throttle valve to the cylinder. By using a steam dome a better quality of steam is obtained, because it is so far above the water level that less water is carried over by the steam into the steam pipe *15*.

This type of boiler has many advantages as well as some disadvantages. It has a large amount of heating surface and it is well distributed. Due to the large amount of heating surface and the excellent draft arrangement, a high evaporation per square foot of heating surface is obtained. It is well adapted to various classes of service and operating conditions. Its disadvantages consist largely in the cost incurred in its maintenance especially in localities where bad water must be used. When this condition is imposed upon it, the flues give trouble by leaking around the joints where they enter the flue sheets *9* and *11*. This leakage may at times become troublesome and in the end costly if proper preventive measures are not taken regularly. Some criticism is also made of this boiler on account of the necessity of using stay bolts in the crown sheet and water legs. It must be admitted that stay bolts are also an item of considerable expense in bad water districts where high steam pressures are used. But by watchful care and manipulation this boiler will give splendid results and for some classes of service it has no equal.

The type of boiler, shown in end view in Fig. 34 and in longitudinal cross section in Fig. 35, is a modification of the well-known and efficient Scotch marine boiler. The boiler consists primarily of a cylindrical fire box *1* enclosed by a circular shell. About midway of the fire box is placed a bridge wall *7*, which deflects the hot gases upward against the shell of the fire box. Ordinary cast-iron grate bars are inserted as at *4*, with the ash pit below. It is to be noted there is a water space *6*, which extends the entire distance around the circular fire box. Above the fire box there are a number of return tubes *3*,

which take the hot gases from the rear end *8* of the boiler to the smoke stack. The path of these gases is indicated by the arrows. To protect the rear sheet from the heat of the gases, a protection plate *9* is riveted or bolted to the plate. As steam is generated it rises, enters the steam dome *12*, passes into the steam pipe *13*, and on to the engine.

It should be noted that this boiler contains no stayed portions and that all the surfaces are circular in form and securely riveted. There being no stayed surfaces the circulation of the water is not interfered with—which is an important consideration—and the opportunity for scale and sediment to collect is greatly reduced, hence there is less likelihood of portions of the boiler becoming heated to the point of injuring the boiler or impairing its safety. Still another feature of interest in the boiler is that the gases are made to traverse the entire length of the boiler twice before being ejected at the stack. This being the case an opportunity is given for a greater portion of the heat contained in the gases to be absorbed by the water, thus securing a higher thermal efficiency than obtained from boilers of the locomotive type. Having no stayed surfaces and a small number of flues results in a small maintenance cost of this type of boiler.

Fig. 34. End View of Modified Scotch Marine Boiler

Traction engines run in sizes from about $7\frac{1}{4}$ inches $\times$ 10 inches to 12 inches $\times$ 12 inches for single engines, and for compound engines the common sizes are $5\frac{3}{4}$ inches $\times$ $8\frac{1}{2}$ inches $\times$ 10 inches to $9\frac{1}{4}$ inches $\times$ 13 inches $\times$ 11 inches. The corresponding horsepower

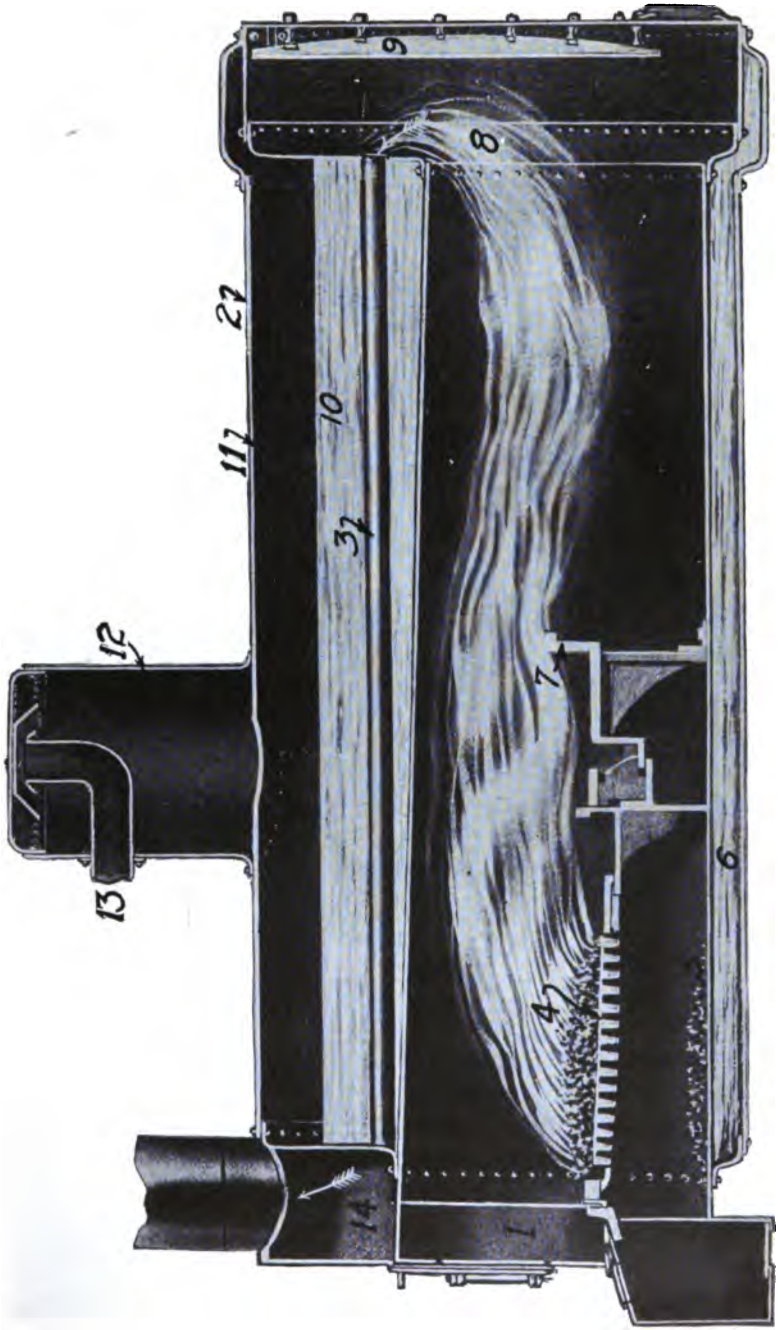


Fig. 35. Longitudinal Cross-Section of Modified Scotch Marine Boiler

developed will run between 15 and 100. The speed attained on the road in miles per hour is about $2\frac{1}{2}$ to 5.

Road Roller Type. The traction engine just considered as an agricultural engine may also be considered as a portable engine or a road locomotive. A portable engine is, therefore, one that can be easily moved about from place to place, or as in the case of the traction engine it may be mounted upon wheels and self-propelled.

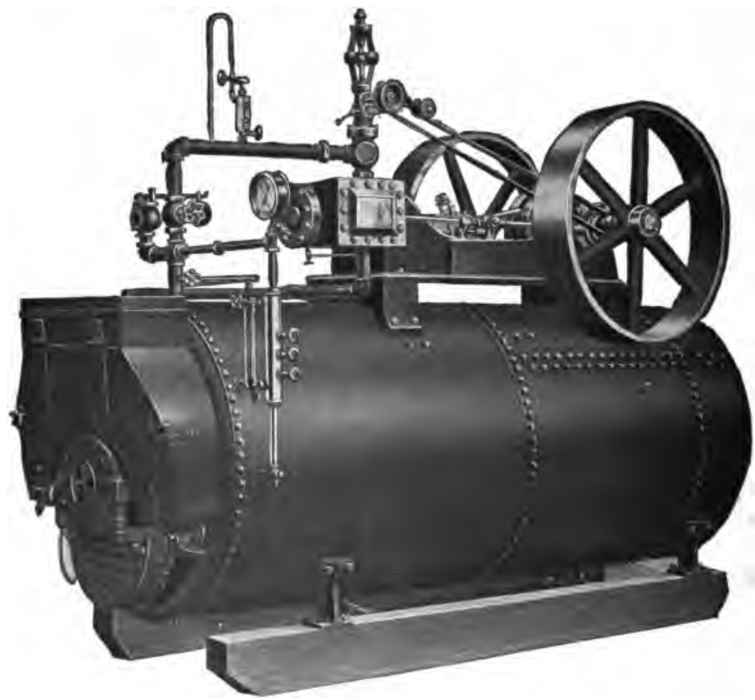


Fig. 36. Semi-Portable Engine and Boiler

Another illustration of a similar type is the ordinary road roller, or road locomotive as it is sometimes called. The principle of its construction and operation is similar to the traction engine, the chief difference between the road roller, or road locomotive, and the ordinary traction engine being that the two front wheels of the traction engine are replaced by a large smooth roller, or cylindrical weight, which revolves as the engine moves. The drive wheels of the road roller are also made large, heavy, and contain no cleats or lugs. These rollers are used in the making of macadamized or other forms of roads.

Semi-Portable Type. The semi-portable engine is usually connected with a small boiler and the two together may be moved from place to place as required. It is not mounted upon wheels but rather on large wood skids, and is moved by being placed either in a wagon or on rollers. It is largely used for hoisting purposes in connection with the construction of large buildings, bridges, etc.

Since the portable and semi-portable engines have no transmission mechanism, they are lighter and considerably cheaper to construct than traction engines.

A very neat, compact, and serviceable type of semi-portable engine is illustrated in Fig. 36. It is mounted upon skids so that it may be easily moved about. The engine is mounted on top of a Scotch marine boiler, similar to the boiler last described, and is of the plain slide valve, center-crank type, with a centrifugal governor. The boiler is equipped with a pressure gauge, water glass, and such other appliances as are usually found in a boiler room of moderate size. The boiler used is sometimes of the locomotive type and, oftentimes, both engine and boiler are of the vertical type. The smaller units are usually of the vertical type, the larger ones of the horizontal type. The semi-portable plant is built in sizes ranging from about 20 to 70 horsepower. If the semi-portable plant, Fig. 36, be mounted on wheels and drawn by horses or some other means, then it is usually classed as a portable engine as distinguished from a semi-portable or traction engine.

LOCOMOTIVE ENGINES

It is not within the province of this work to fully discuss the modern railway locomotive, but suffice it to say that no other power-developing unit has been so rapidly developed with such economical results. Considering the exacting demands made upon a locomotive, its performance is remarkable. The locomotive consists of two primary elements, namely, the boiler which generates the steam and the engines which convert the energy of this steam into useful work by giving motion to the transmission mechanism.

Boiler. Fig. 37 illustrates a modern locomotive boiler. It consists of a cylindrical barrel and an enlarged rear end which contains the fire box. The fire box is securely fastened in the boiler shell by stay bolts and radial stays. A few rows of sling stays are sometimes

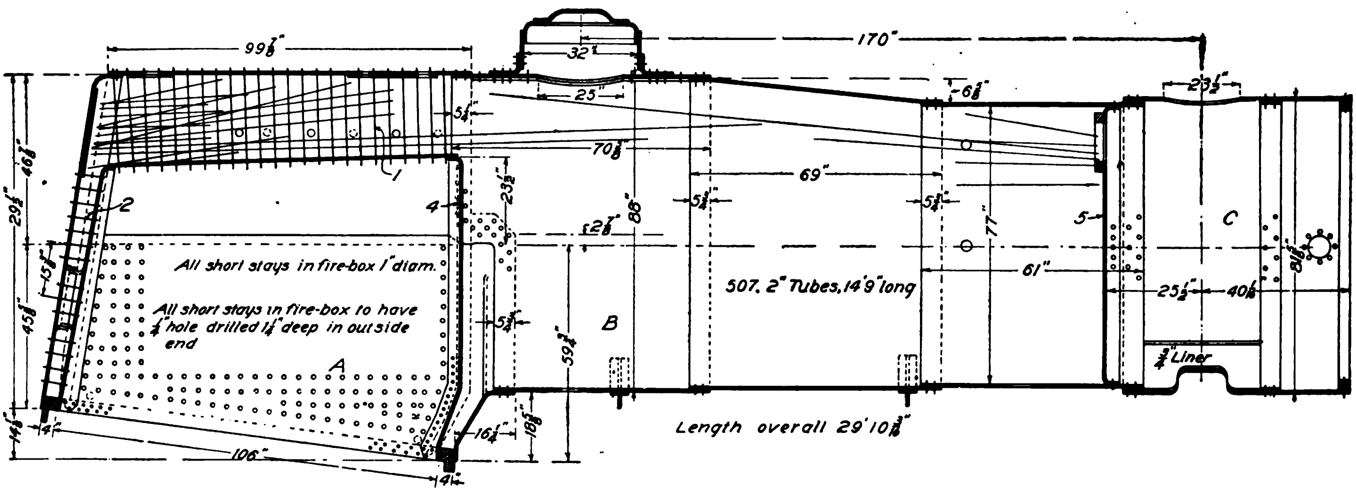


Fig. 37. Detailed Drawing of Modern Locomotive Boiler

used at the front end of the fire box to allow for expansion and contraction of the sheets. The boiler is divided into three distinct departments, as the fire box *A*, the water space *B*, and the smoke box *C*. The sheets *4* and *5*, which separate these departments, are known as the back and front flue sheets, respectively. The flue sheets are drilled with holes to receive the flues.

Flues. In the particular boiler illustrated about 400 2-inch flues are used. These flues extend from flue sheet to flue sheet and form a passage for the gases to travel from the fire box to the smoke box. Surrounding the flues in the space *B* and surrounding the fire box is water, which is vaporized into steam due to the combustion of fuel in the fire box. The total amount of heating surface will vary from 2,500 to over 4,000 square feet, according to the type and size of the locomotive. Of this total amount of heating surface only a very small per cent is furnished by the fire box, there being usually only about 200 square feet of heating surface contained in the fire box. It is evident, therefore, that the flues are a very important part of the locomotive boiler.

Grate Area. The amount of grate area varies from about 40 to 60 square feet. It must be obvious that in order for so small a grate area to supply sufficient heat to such a large amount of heating surface there must be a very high rate of coal consumption per square foot of grate area. A series of tests made at St. Louis during the Exposition in 1904 demonstrated that the amount of dry coal fired per square foot of grate area per hour varied from 20 to as high as 130 pounds. These results were obtained from several different types of locomotives operated under widely different speeds and loads, hence the above figures may be taken as approximating the maximum and minimum consumption under ordinary running conditions. Under these very widely different operating conditions it was found that the equivalent evaporation per pound of dry coal varied from $6\frac{1}{2}$ to 12 pounds, which compares very favorably with stationary boiler performance which gives an average evaporation of about 8 pounds of water per pound of coal.

Mechanical Efficiency. The mechanical efficiency of a locomotive is also very good. Through a long series of tests conducted on a well-equipped locomotive testing plant, a mechanical efficiency of 65 to 85 per cent was obtained. The same degree of efficiency has

been obtained in various other tests and under more adverse conditions. The locomotive is also very efficient in the use of steam. The St. Louis tests showed that simple freight locomotives gave an average minimum water consumption per indicated horsepower per hour of 23.67 pounds. The water consumption per indicated horsepower per hour under maximum load was 23.83 pounds, whereas the maximum rate was 28.95 pounds. For compound freight locomotives the average steam consumption was: minimum load 20.26 pounds, maximum load 22.03 pounds, and maximum consumption 25.31 pounds. The average steam consumption for simple passenger locomotives was: minimum load 18.86 pounds, maximum load 21.39 pounds, and maximum consumption 24.41 pounds. When these figures are compared with those of the best stationary engines, some idea of the economy of the locomotive can be obtained. The steam consumption of an automatic, tandem-compound, condensing stationary engine with piston valves under full load is about 18 pounds per indicated horsepower per hour, whereas the compound non-condensing locomotive is about 21 pounds. A Corliss engine or a medium speed, four-valve simple engine will give a minimum steam consumption of about 22 pounds per indicated horsepower per hour under full load. A simple freight engine under full load will use about 23.5 pounds of steam per indicated horsepower per hour. The foregoing figures speak well in favor of the economy of a steam locomotive, which is operated under conditions unfavorable to the securing of good economy.

Engine Characteristics. The engines used on locomotives may be simple or compound; in fact, both are used extensively, although the simple type predominates. It is to be noted that the steam locomotive is equipped with two separate and distinct engines—one being attached to each side of the boiler, and both attached to the driving wheels through the medium of the frames, etc.

The mechanical construction of these engines is quite similar to that of the type already described in this work. Certain features are made necessary in order to properly tie together the engine, boiler, and transmission mechanism. Perhaps the most noticeable change in detail is in the construction of the cylinders and valve seats, otherwise there is little variation from the well-established principles of engine design. The valves, rods, crossheads, guides, etc., are

made of the same high-grade material and constructed in the same first-class manner as is required for a good stationary engine. This being true, much of the discussion of the steam stationary engine and its parts already given is applicable to the engines of a locomotive. There are, however, many perplexing questions that arise with reference to the performance and operation of the locomotive as a whole that are never encountered in stationary practice, due to the unusual and sometimes trying conditions under which the locomotive must be operated. The solution of these problems demands a great amount of ingenuity and engineering ability.

To discuss the various types of locomotives and tell the many interesting and important points connected therewith would require entirely too much space, so the discussion must be confined to narrow limits.

Types of Locomotives. There are certain types of locomotives common in American practice which have special names. The eight-wheel or "American" passenger type of locomotive has four coupled driving wheels and a four-wheeled truck in front. The "ten-wheel" type has six coupled drivers and a leading four-wheel truck. This type is used for both freight and passenger service. The "Mogul" type is used altogether for freight purposes; it has six coupled drivers and a two-wheel or pony truck in front. The "Consolidation" type is used for heavy freight service. It has eight coupled drivers and a pony truck in front. There are also a great many special types for special purposes. In switch yards a type of engine is used which has four or six drivers with no truck. The Forney type has four coupled driving wheels under the engine and a four-wheel truck carrying the water tank and fuel. This type is used on elevated roads largely. "Decapod" engines are a type used for heavy freight service, having ten coupled driving wheels and a two-wheel truck in front. A tank engine is one which carries the feed water in tanks on the engine itself instead of in the tender, as in other engines.

A locomotive of modern design that is being largely used for fast freight service and for heavy passenger service is illustrated in Fig. 38. It is commonly known as the Atlantic type locomotive, having four leading truck wheels, four coupled drivers, and a two-wheel trailing truck. The leading truck wheels serve in a guiding

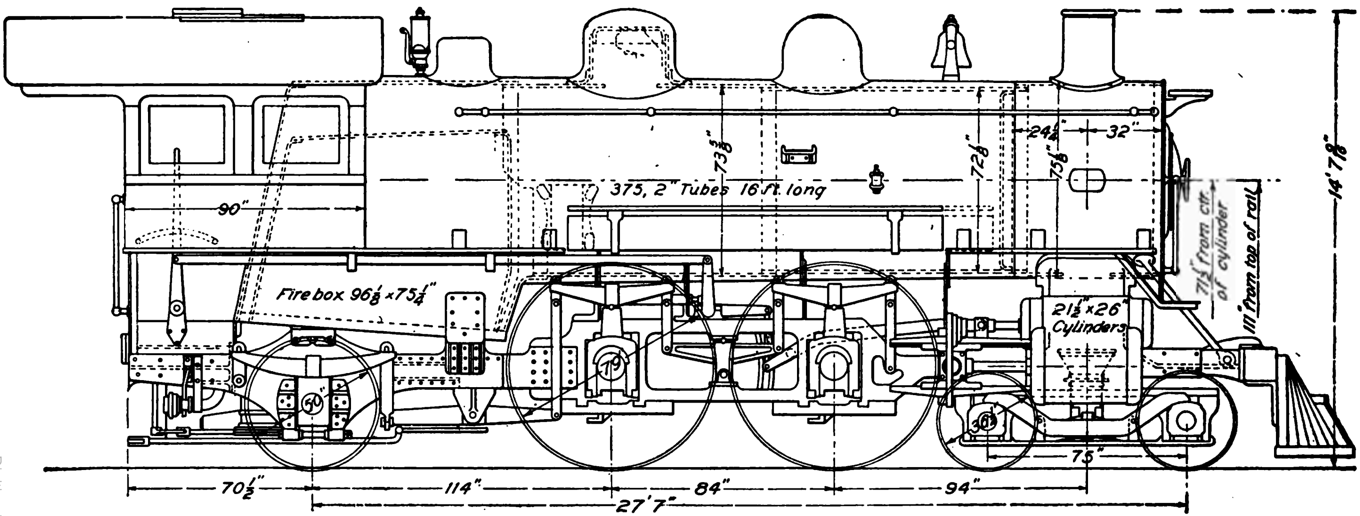


Fig. 38. Detailed Drawing of Modern Locomotive Engine of the "Atlantic" Type

capacity. This engine is a compound type with piston valves, is well designed, is neatly proportioned, and admirably fulfills every requirement.

Compound Type. In connection with the subject of compounding just mentioned it may be said that in recent years the compound locomotive has been found in increased numbers on American railroads. A type of compound that has given especial good service and which is being adopted by many roads for heavy hill climbing duty is the Mallett Articulated Compound. The adoption of the compound locomotive has been due to a general opinion among railroad officials that the findings of a committee of the American Master Mechanics Association were true, as demonstrated by practice. This Committee says of compounding:

(a) It has achieved a saving in the fuel burned, averaging 18 per cent at reasonable boiler pressures.
(b) It has lessened the amount of water to be handled.
(c) The tender can, therefore, be reduced in size and weight.
(d) It has increased the possibilities of speed beyond sixty miles per hour, without unduly straining the engine.
(e) It has increased the haulage power at full speed.
(f) In some classes of engines it has increased the starting power.
(g) It has lessened the valve friction per horsepower developed.

A number of other reasons are given in their report. Notwithstanding these facts, however, the compound locomotive has not come into very general use on railroads.

WATER PUMPS

The subject of pumping engines is a very broad one, and one which has received the thought and study of the most eminent engineers for many decades. From the earliest history of man there is gleaned the fact that human ingenuity and skill had been devoted in those early times to the perfection of some kind of power pump. It would be a difficult matter to mention an industry of any character or description but what a pump was needed somewhere in the enterprise. It was first used in a large way in the mining industries for pumping water out of the mines. Today it is found in all power houses, mines, and factories of various kinds. Both the large and small cities depend upon it for their water supply. The heating and ventilating systems of modern apartment houses and office

buildings use the pump, and mention might be made of many other instances where the water pump is indispensable.

There are two general classes of pumps, namely, crank or flywheel type and direct acting pumps.

Crank or Flywheel Type. The crank or flywheel type was the first form to be developed. These pumps vary greatly both in their design and in the details of their construction. They are of varying sizes, including some of the largest and most expensive in the world. As a general thing they are used in heavy hydraulic enterprises, for furnishing water supply for cities, and in various other enterprises where a large and constant supply of water is demanded. In this class of pumps or engines the application of the power in the steam cylinders in driving the pump plunger or piston varies greatly both in design and detail of construction. Long or short beams or bell cranks may be used and sometimes gearing may be employed, but in all cases the limit of the stroke of the steam piston and of the pump plunger is governed by the crank of a revolving shaft. In pumping engines it is not absolutely necessary to have a revolving shaft, the only requirement being that the piston in the pump cylinder shall be driven back and forth with a plain reciprocating motion which may be exactly like that of the steam piston. For this reason, in early pumping engines and also in modern engines, the reciprocating motion of the steam piston is applied directly, or through a beam, to produce the reciprocating motion of the pump piston or plunger without the use of any revolving part. Frequently, however, it is desirable to use a flywheel so that the steam may be used expansively, and in these cases, of course, a revolving shaft must be used.

Cameron Belt-Driven Pump. The power pump used as an illustration, Fig. 39, is a belt-driven one. The belt is placed on the pulley *1* and can be shifted to a loose pulley by the shifter *2*, when desired. The shaft *4*, which is driven by the belt pulley, extends across the frame and has attached to it a flywheel *5* and a small gear wheel, which meshes with the large gear wheel *3*. The gear wheel *3* is keyed to the crank shaft *6*, hence, when it is driven, the crank shaft is made to revolve, which in turn gives a back and forth movement to the piston as in the ordinary steam engine. The flywheel *5*, attached to the revolving shaft, may be of greater or less diameter and weight, depending on the condition under which the pump is to be operated.

In addition to assisting the crank to pass the dead center at each end of the stroke of the piston, it can be employed as a reservoir in which any excess energy may be stored at the beginning of each stroke and drawn out during the latter part of the stroke, where the force of the water column is greater than that of the steam. By this means it is possible to use shorter cut-offs in the cylinder than could otherwise be permitted; hence, a resulting saving in steam. Many means may be used to drive the power pump. While the illustration shows

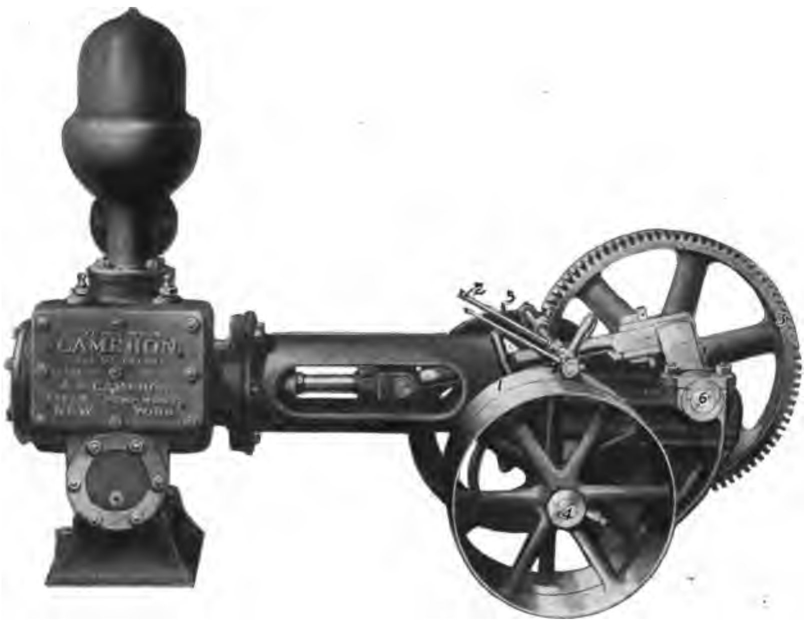


Fig. 39. Belt-Driven Power Pump

one belt driven, yet they are frequently electrically driven, and sometimes the revolving shaft is attached to the shaft of a gas or steam engine.

Deep-Well or Mine Pump. For deep-well or mine pumping, the cylinders are often set in a vertical position directly over the pump cylinder. The piston rod extends from the steam cylinder directly below to the pump plunger. Sometimes it is possible to use steam expansively in these pumps by reason of the weight of the reciprocating parts. When the weight is sufficient, the steam can be cut off before the end of the stroke and the momentum of the parts will

be enough to just finish the stroke, consequently these pumps are sometimes compounded. They are used only in pumping from very deep wells.

Direct-Acting Type. A direct-acting steam pump is one in which there are no revolving parts, such as shafts, cranks, and fly-wheels, the power of the steam in the steam cylinder being transferred to the piston or plunger in the pump in a direct line through the use of a continuous rod or connection. In pumps of this construction there are no weights in the moving parts, other than those required to produce sufficient strength in such parts for the work they are required to perform and, as there is consequently no opportunity to store up power in one part of the stroke to be given out at another, it is impossible to cut off the steam in the steam cylinder during any

Fig. 40. Direct-Acting. Duplex Pump with Rocker and Bell-Crank Lever

part of its stroke. The uniform and steady action of the direct-acting steam pump is dependent alone on the use of a steady uniform pressure of steam through the entire stroke of the piston, against a steady, uniform resistance of water pressure in the pump; the difference between the power exerted in the steam cylinders and the resistance in the pump governs the rate of speed at which the piston or plunger of the pump will move. The length of the stroke of the steam piston within the steam cylinders of this class of pumps is limited, and is controlled alone by the admission, compression, and release of the steam used in the cylinders.

Duplex Pump with Rocker and Bell-Crank Lever. The direct-acting steam pump, Fig. 40, is known as a duplex pump and consists simply of two direct-acting steam pumps placed side by side. The steam pistons are at one end and the water pistons at the other.

The steam pressure acts directly on the pistons; no flywheel is used; and since the reciprocating parts are comparatively light and there is no revolving mass to carry by the dead points, it is evident that in the ordinary form there can be no expansion of steam. The pump is inexpensive and gives a positive action. It uses a relatively large quantity of steam, but for small work the absolute amount is not very great.

On the piston rod of each pump is a bell-crank lever which operates the valve of the other pump. There must be a rocker on one side and a bell-crank lever on the other, because of the relative motion of the valves and pistons. The first piston, as it goes forward, must use a rocker, because it draws the second valve back. The second piston, as it goes back, must use a bell-crank lever because it must push the first valve back in the same direction as its own motion. The two pistons are made to work a half-stroke apart, thus one begins its stroke when the other is in the middle. In this way a steady flow of water is obtained, as both pumps discharge into the same delivery pipe. In large pumps of this kind, and even in some small ones, the motion described above merely admits steam to a small auxiliary piston on each steam cylinder, which then moves the main steam valve by steam pressure.

Duplex Pump with Tappet. Some pumps operate the steam valve by means of a tappet instead of a rocker and a bell-crank lever, Fig. 41. Its construction and operation is as follows:

A is the steam cylinder; *C*, the piston; *L*, the steam chest; *F*, the chest plunger, the right-hand end of which is shown in section; *G*, the slide valve; *H*, a lever, by means of which the steam-chest plunger *F* may be reversed by hand when expedient; *II* are reversing valves; *KK* are the reversing valve chamber bonnets; and *EE* are exhaust ports leading from the ends of the steam chest direct to the main exhaust and closed by the reversing valves *II*.

The piston *C* is driven by steam admitted under the slide valve *G*, which, as it is shifted backward and forward, alternately connects opposite ends of the cylinder *A* with the live steam pipe and exhaust. This slide valve *G* is shifted by the auxiliary plunger *F*, the latter having hollow ends which are filled with steam, and this, issuing through a hole in each end, fills the spaces between it and the heads of the steam chest in which it works. Pressure being equal at each

end, this plunger *F*, under ordinary conditions, is balanced and motionless; but when the main piston *C* has traveled far enough to strike and open the reverse valve *I*, the steam exhausts through the port *E* from behind that end of the plunger *F*, which immediately shifts accordingly and carries with it the slide valve *G*, thus reversing the pump. No matter how fast the piston may be traveling, it

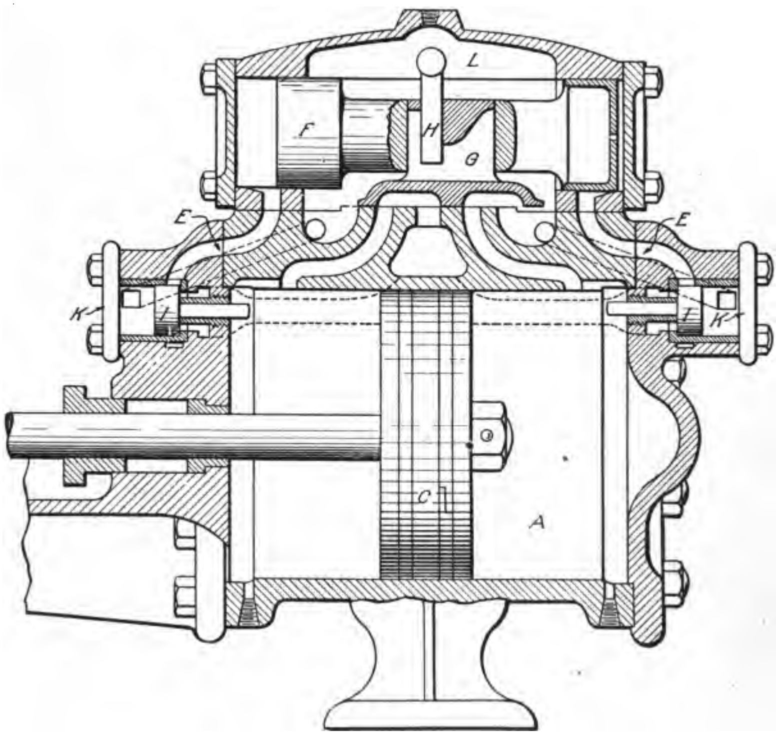


Fig. 41. Section of Pump Cylinder Showing Valve Operated with Tappet

must instantly reverse on touching the valve *I*. In its movement the plunger *F* acts as a slide valve to close the port *E* and is cushioned on the confined steam between the ports and steam-chest cover. The reverse valves *II* are closed as soon as the piston *C* leaves them by a constant pressure of steam behind them conveyed direct from the steam chest through the ports shown by dotted lines.

The motion of the piston *C*, Fig. 42, is transmitted through the rod *M* to the water piston in the cylinder *R*. As the piston moves

back and forth, water enters through the intake valves *O* and leaves through the discharge valves immediately above, and finally leaves through the delivery pipe *P*. In order to create a more continuous flow of water, an air chamber *Q* is provided. Any sudden variation in the pressure in the line is taken up largely by the air chamber. It also serves to lessen the effect of water hammer.

MARINE ENGINES

Beam Type. The first steam vessels were fitted with paddle wheels, and as beam engines were the most common, this form of

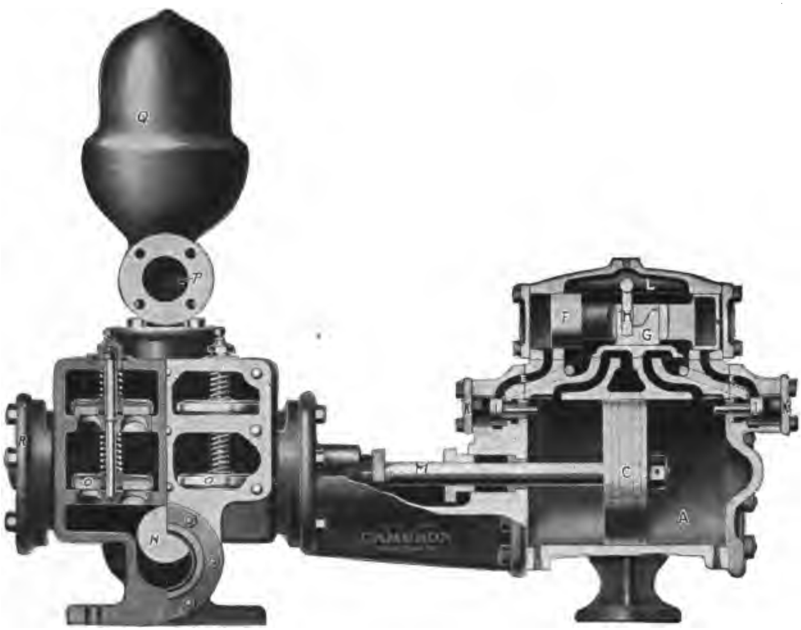


Fig. 42. Section of Duplex Pump with Tappet

engine was used. Its construction, however, was somewhat modified for this service. This arrangement of beam engine and paddle wheel was used for many years and was applied to ocean vessels as well as to small river boats. It is still used, especially in this country, on river steamers and some coast steamers. The beam is supported by a large A-frame on the deck, and the engines are about on a level with the shaft.

Engines of this type take up rather more room than those now in common use, partly because of great size, and also because of the shaft and paddle wheels. Another disadvantage is that in heavy weather when one paddle wheel is thrown out of the water the other is deeply immersed and takes all the strain, so that there is a tendency to rack the boat. Then again if the boat is loaded heavily, the paddle blades are very deeply immersed; while if light, they barely touch the water. It is difficult to handle the engines satisfactorily under either condition.

Inclined Type. The introduction of the screw propeller overcame these difficulties very largely and at the same time required a high speed engine. At first, the increased speed was supplied by the use of spur-wheel gearing, but gradually higher speed engines were built and connected directly to the propeller shaft. It was, of course, difficult with small width at each side of the shaft to use horizontal engines, therefore various arrangements of inclined engines were used before the vertical engine was finally chosen by all as the standard form for marine work. It is only in recent years that the vertical engine has become general in naval work and in merchant steamers.

Vertical Type. In merchant ocean steamers the common form has three cylinders set in line, fore and aft, above the shaft, the cranks being set 120 degrees apart in order to give a more even turning moment. The three cylinders are worked triple expansion, the valves being usually of the piston type on the high and intermediate and double-ported slide type on the low. Sometimes piston valves are used on all the cylinders. Plain slide valves are not suitable for high pressure work of any kind.

Surface Condensers Necessary for Ocean Vessels. For engines on ocean vessels it is necessary to use surface condensers in order that the same water may be used over and over again. If it were necessary to take in sea water for the boilers, they would soon become clogged with the salt and require cleaning. Surface condensers for marine work are generally made up of a large number of brass tubes from $\frac{3}{4}$ inch to 1 inch in diameter. In some cases the cold water is forced to flow through the tubes while the steam comes in contact with the outside of the tubes.

In any marine plant there are four special pumps. The first is the *air pump* for the condenser. This is usually made large so

that in case there is a leak in the condenser it can take charge of the water even if it becomes necessary to run as a jet instead of a surface condenser. The second is the *feed pump* for the boilers. The third is the *circulating pump,* which forces the current of cold water through the condenser. The fourth is the *bilge pump,* which pumps out water that gathers in the bilge of the ship by leakage or otherwise. In case of a serious leak, all the pumps can be made to pump from the bilges. In some old types all of these pumps were worked from the main engine but in modern marine plants the feed pump and the circulating pump are generally separate, as is also the bilge pump. The circulating pump is, in many modern engine rooms, of the centrifugal type.

SPECIAL ENGINES

Under this heading may be placed a large number of engines which have been built for a very definite field of usefulness, such as various types of fire engines and automobile engines, where steam is used as the motive force. Again a number of experimental engines have been built, commonly known as freak engines, having peculiar construction and design, which never got beyond the experimental stage. Rotary engines as well as rotary pumps have been used to some extent, but the rotary engines thus far developed have been so extravagant in steam consumption that their use has been discontinued. It is thus seen that under the head of special engines many of the engines already discussed, as well as an untold number of others of more or less merit, may be properly classed.

The special engines referred to above were not mentioned for the purpose of studying them, but rather to indicate that outside and distinct from the steam engines classified and considered, there are a large number of special types that should not be entirely ignored.

MECHANICAL AND THERMAL EFFICIENCY

The brief historical review and the study of the various types of engines have served to unfold the degree of perfection that has been attained in the design and details of construction of the modern steam engine. From a mechanical standpoint, the modern engine is highly efficient. A mechanical efficiency, that is,

$\frac{\text{Brake horsepower}}{\text{Indicated horsepower}}$, of from 85 to 95 per cent is not infrequently obtained. An actual test of a 12-inch × 19½-inch × 15-inch tandem-compound Corliss engine operating non-condensing gave a mechanical efficiency of 94 per cent. That is to say, if the engine was developing 120 horsepower in the cylinders, that 112.8 horsepower would be delivered by the engine to the flywheel. In other words, the horsepower used in overcoming the friction of the various moving parts was only 7.2 or 6 per cent of the total horsepower developed.

Low Thermal Efficiency Inherent. From the standpoint of thermal efficiency, however, the modern engine is very inefficient, but it is much more efficient than the older types. Even the maximum thermal efficiency obtained is only about 15 per cent, and, under favorable conditions, this very low figure may be so reduced that the engine is operated at a great economic loss. It is now proposed to briefly point out some of the causes for the very low thermal efficiency obtained and to indicate some of the means that have been employed to increase the thermal output of the steam engine. In order to make this study it becomes necessary to again refer to steam and its properties. It is well known that steam contains a great deal of heat, and that this heat can be converted into useful work by allowing the steam to pass from the high temperature of the heat generator to the lower temperature of the refrigerator, during this change giving up heat. There are several forms of heat engines, all of which convert the heat contained in some substance into work. The theoretically perfect engine shall be considered first, and after that the modifications that go to make up the steam engine of today.

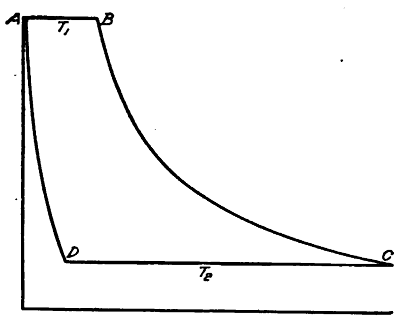


Fig. 43. Theoretical Indicator Diagram

Ideal Engine. The theoretical engine (see Fig. 43), is supposed to receive heat from the generator at constant temperature T_1 until communication is interrupted at B. The working substance expands

to C without losing or gaining any heat from external sources until the temperature of the refrigerator is reached. The engine now rejects heat at the constant temperature T_2 of the refrigerator and then compresses the working substance without loss or gain in the quantity of heat until the temperature of the heat generator is reached. These are ideal conditions and, if fulfilled, the efficiency of the perfect engine will depend only on the difference between the temperature at which heat is received and rejected or, in other words, it depends only upon the difference in temperature between the generator and the refrigerator.

If T_1 equals the absolute temperature of the heat received and T_2 equals the absolute temperature of the heat rejected, then the thermal efficiency E of the engine will be represented by the formula

$$E=\frac{T_1-T_2}{T_1}$$

Or, in other words, the efficiency equals the absolute temperature of the heat rejected, subtracted from the absolute temperature of the heat received, and the remainder divided by the absolute temperature of the heat received.

EXAMPLE. Given an engine using steam at 120 pounds absolute pressure, and exhausting at atmospheric pressure. What is the thermal efficiency?

SOLUTION. The absolute temperature corresponding to 120 pounds pressure is 341.05+461, or 802.05°, and the absolute temperature of the exhaust is 212+461, or 673°. Then

$$\begin{aligned} E&=\frac{802.05-673}{802.05} \\ &=.16, \text{ or 16 per cent} \end{aligned}$$

Losses in Practical Engine. *In General.* In actual engines this efficiency can not be realized, because the difference between the heat received and the heat rejected is not all converted into useful work. Part of it is lost by radiation, conduction, condensation, leakage, and imperfect action of the valves. The cylinder walls of the theoretical engine are supposed to be made of a non-conducting material, while in the actual engine the walls are of metal, which admits of a ready interchange of heat between cylinder and steam. This action of the walls can not be overcome and is so important that a failure to consider its influence will lead to serious errors in

computations, and no design can be made intelligently if based on the theory of the engine with non-conducting walls. In theoretical engines steam expands without the loss of any heat, while in the actual engine a large amount of heat is lost by radiation. There is also a considerable loss of pressure between the boiler and the engine, due to the resistance offered by the pipes and cylinder passages. In a slow-speed engine with large and direct ports and valves this trouble is reduced to a minimum. The imperfect action of the valve gears may also be lessened with due care, but the action of the cylinder walls still remains to be overcome.

Theoretical and Actual Card Analyses. In the theoretical card, admission is at constant boiler pressure, cut-off is sharp, expansion is complete—that is, expansion continues until the temperature falls to that of the condenser and the exhaust is at condenser pressure—and the piston always sweeps the full length of the cylinder.

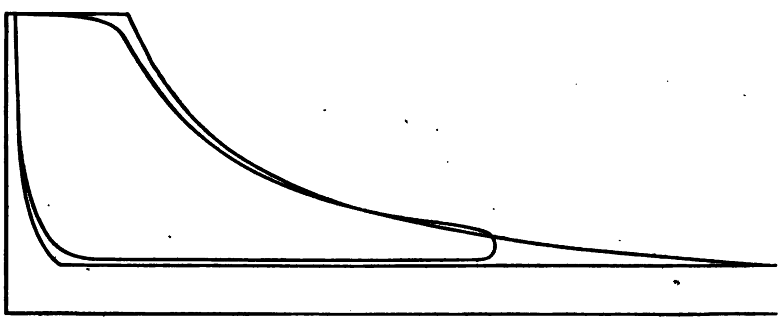


Fig. 44. Superposed Ideal and Actual Indicator Diagrams

In the actual engine there is a considerable loss of pressure between boiler and engine, and the wire drawing of the ports and valves tends to cause a sloping steam line. Condensation at the beginning of the stroke causes the real expansion line to fall below the theoretical, while re-evaporation causes it to rise above the theoretical toward the end of expansion. In the actual engine, release takes place before the end of the stroke, expansion is not complete, that is, the pressure at release is above that of the condenser, and the resistance of exhaust ports causes the back pressure to be above the actual condenser pressure. Moreover, the piston does not sweep the full length of the cylinder, and the clearance space must be filled with

steam, which does very little work. The theoretical and actual cards are shown in Fig. 44.

Mechanical Losses. It has been shown that the efficiency of the theoretical engine is purely a thermal consideration; the efficiency of the actual engine, however, is largely a mechanical matter. The unit of work is the horsepower, which corresponds to the development of 33,000 foot pounds per minute. As 778 foot pounds are equivalent to one British Thermal Unit, 33,000 foot pounds per minute, or one horsepower, is equivalent to $33000 \div 778$ or 42.42 British Thermal Units. Now if a certain engine uses 84.84 British Thermal Units per horsepower per minute, it is evident that its efficiency will only be one-half, or 50 per cent, because 42.42 is one-half of 84.84. Hence, it may be said that the efficiency of the actual engine is equal to $\dfrac{42.42}{\text{British Thermal Units per horsepower per minute}}$.
This efficiency is always much less than that of the perfect engine.

ANALYSIS OF LOSSES

The effect of some of the losses in the steam engine and the methods for decreasing them will now be considered.

Radiation. In the first place, the metal walls of the cylinder, being good conductors of heat, become heated by the steam within and transmit this heat by conduction and radiation to the air or external bodies. With the cylinder well lagged, much less heat is lost by radiation. If the lagging were perfect and the temperature of the cylinder remained the same as the temperature of the steam throughout the stroke, there would be no loss by radiation, but heat would still be lost by conduction to the different parts of the engine.

Cooling by Expansion. During expansion, the temperature and pressure of the steam decrease as the volume increases, and the temperature at exhaust is much less than the temperature at admission. In the perfect engine, the working substance after exhaust is compressed to the temperature at admission, but in the actual engine much of this steam is lost and the compression of a part of it is incomplete, so that its temperature is less than the temperature at admission.

Steam Condensation and Re-Evaporation. Consider an engine operating with admission at 100 pounds absolute and exhaust at 18 pounds absolute. From steam tables the temperature at admission is found to be 327.6°, and at exhaust 222.4°. The metal walls of the cylinder, being good conductors and radiators of heat, are cooled by the low temperature of exhaust, so that the entering steam in passing through ports and into a cylinder is subjected to a temperature more than 100° cooler than the steam. This means that heat must flow from the steam to the metal until both are of the same temperature. This causes the steam to give up part of its latent heat, and as saturated steam can not lose any of its heat without condensation, the cylinder walls become covered with a film of moisture, usually spoken of as initial condensation. This condensation in simple unjacketed engines, working under fair conditions, may easily be 20 per cent or more of the entering steam. The moisture in the cylinder has, of course, the same temperature as the steam; it has simply lost its heat of vaporization.

Although metal is a good conductor of heat it can not give up or absorb heat instantly; consequently during expansion, the temperature of the steam falls more rapidly than that of the cylinder. This allows heat to flow from the cylinder walls to the moisture on them. As fast as the steam expands so that the pressure in the cylinder becomes less, this condensation will begin to evaporate. As the pressure falls it requires less and less heat to form steam and, therefore, more and more of this moisture will be evaporated. At release the pressure drops suddenly, more heat at once flows from the cylinder walls, and re-evaporation continues throughout the exhaust. Probably all of the water remaining in the cylinder at release is now re-evaporated, blows out into the air of the condenser, and is lost as far as useful work is concerned.

The steam that is first condensed in the cylinder does no work; its heat is used to warm up the cylinder, and later, when it is re-evaporated, it works only during a part of the expansion and at a reduced efficiency, because it is re-evaporated at a pressure and, consequently, at a temperature very much lower than that of admission. If the cut-off is short, perhaps 20 per cent of the steam condensed may be re-evaporated during expansion; if the cut-off is long, 10 per cent may be re-evaporated, the rest remaining in the cylinder at release,

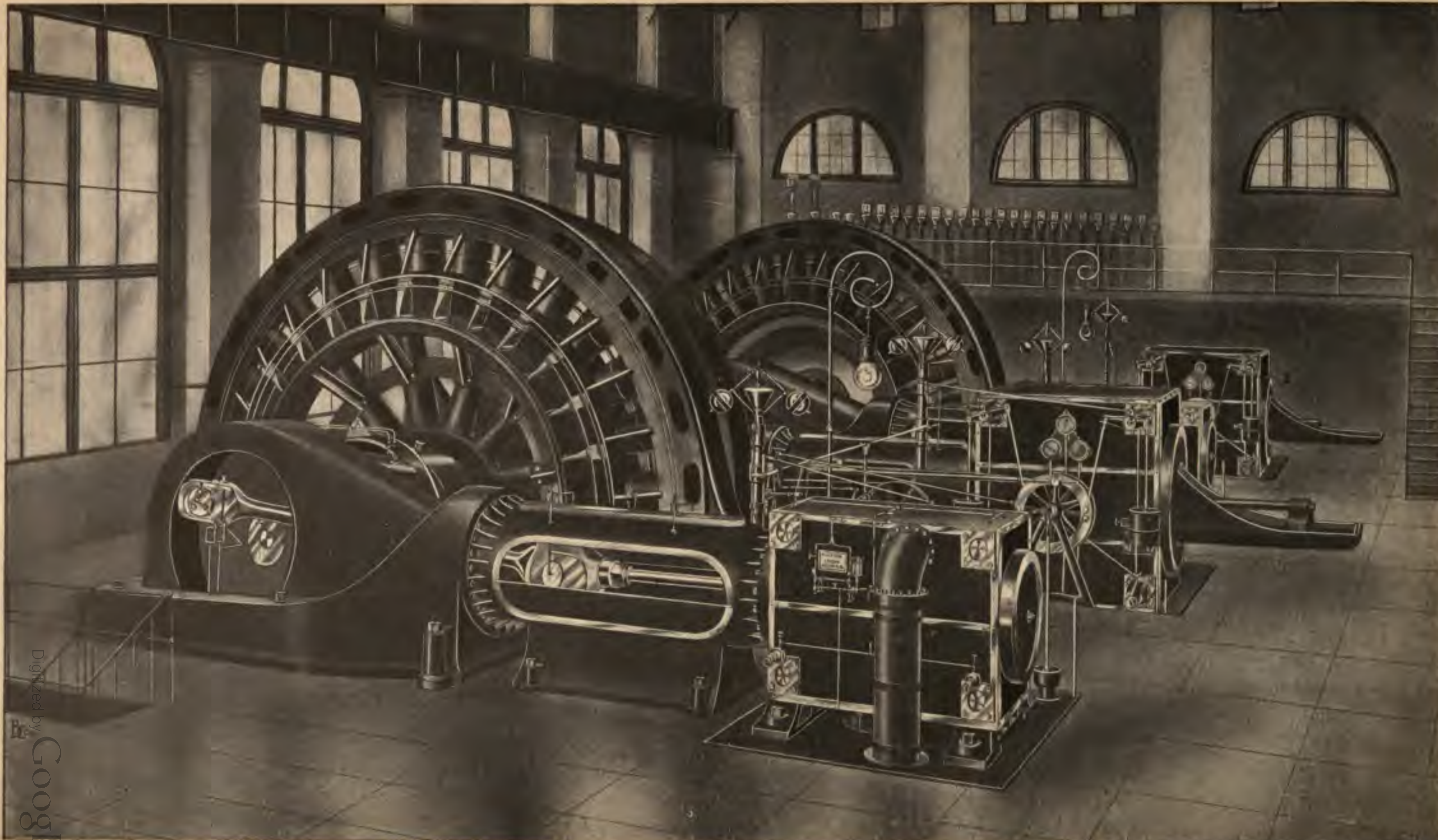
still in the form of moisture. Thus some of the entering steam passes through the cylinder as moisture until after cut-off, and still more passes entirely through without doing any work.

Suppose an engine is using 30 pounds of steam per horsepower per hour and admission is at 100 pounds absolute. The latent heat of vaporization at this pressure is 884 British Thermal Units per pound. If the condensation amounts to $33\frac{1}{3}$ per cent, then 10 pounds are condensed and there is lost 10 times 884, or 8,840 British Thermal Units per hour, or 147.3 per minute; and since 42.42 British Thermal Units represents 1 horsepower, there is lost by condensation 147.3 divided by 42.42, or $3\frac{1}{2}$ horsepower (nearly). If the cut-off is shortened, the condensation increases and may amount to 50 per cent. Of course very much less steam is used at a short cut-off than with a long cut-off, and doubtless in many cases 50 per cent of the steam at short cut-off is not as great an absolute quantity as 30 per cent at a long cut-off.

Exhaust Waste. In addition to the actual loss from condensation in the cylinder, there is still another loss due to re-evaporation. Suppose, as before, that 10 pounds of steam are condensed in the cylinder, and that 20 per cent of this is re-evaporated during expansion. This will leave 8 pounds to be re-evaporated during exhaust. Suppose the exhaust is at 3 pounds above atmospheric pressure, or 18 pounds absolute (about). Then the heat of vaporization is 958.5 British Thermal Units per pound of steam, and it will require 8 times 958.5, or 7,668.0 British Thermal Units, to evaporate the 8 pounds. All of this heat is taken from the cylinder, leaving the engine much cooler than it would be were it not for this re-evaporation. This gives some idea of the great amount of heat passing away at exhaust, which is known as the *exhaust waste.*

Clearance. In all cylinders it is necessary to have a little space between the cylinder cover and the piston when at the end of the stroke. In vertical engines the space is greater at the bottom than at the top. The volume of this space, together with the volume of the steam ports, is called the clearance. It varies from 1 to about 15 per cent, depending upon the type and speed of the engine—the higher the speed, the greater the clearance. This clearance space must be filled with steam before the piston receives full pressure; and the volume of the clearance offers additional surface for condensation.

Friction. Another important loss is that due to friction. It is well known that it takes considerable power to move an unloaded engine; if fitted with a plain, unbalanced slide valve, the power necessary to move the valve alone is considerable. The piston is made steam-tight by packing rings, and leakage around the piston rod is prevented by stuffing boxes. All these devices cause friction as well as wear at the joints. The amount of power wasted in friction varies greatly, depending upon the kind of valves, general workmanship, state of repair, and lubrication.



FULTON CORLISS HEAVY-DUTY CROSS-COMPOUND CONDENSING ENGINES, DIRECT-CONNECTED TO 2,250-K. W. RAILWAY GENERATORS
Fulton Iron Works, St. Louis, Mo.

STEAM ENGINES

PART II

OPERATION ECONOMIES

The foregoing discussion has served to indicate that the larger part of the heat loss occurring in the steam engine is due to initial condensation, exhaust waste, and clearance, although the effect of the latter has been greatly reduced by improvement in design. Regarding the methods devised for reducing the amount of initial condensation, the high speed engine has in a measure decreased this difficulty because of the very high piston speed employed. Since the piston speeds are high, the length of time the steam remains in the cylinder has been greatly lessened; hence the transference of heat is considerably reduced. The piston speed is limited, however, by the performance of the valve gear, it being well known that the most efficient valve gears are those employed on the low speed engines. Increased piston speed also calls for more clearance space, hence the possible gain in economy from high piston speed is limited by the performance of the valve gear and the clearance required for the higher speeds.

The application of the idea of multiple expansion, or compounding, has materially reduced the losses both by lessening the amount of condensation and also by utilizing the re-evaporated steam and the steam that leaks by the piston, which in some cases may be considerable, and this important improvement will be discussed first. In addition other means have been employed for the purpose of increasing the economic performance of the steam engine, as for instance, *jacketing*, *superheating*, and the *use of condensers*.

MULTIPLE EXPANSION

Two engines may be used together on the same shaft, partly expanding the steam in one of the cylinders and then passing it over to the other to finish the expansion. One advantage from this

arrangement is that the parts can be made lighter. The high pressure cylinder can be of much less diameter than would be possible if the entire expansion were to take place in one cylinder. This, of course, makes the pressure exerted on the piston rod much less, and the piston rod and connecting rod can thus be made much lighter. The low pressure cylinder must be larger than it otherwise would be, but its parts need not be much heavier, because the pressure per square inch is always low.

This arrangement gives not only the advantage of lighter parts, but a decided increase of economy over the single cylinder type. If attention is given to the matter, a loss of economy would be expected, because the steam is exposed to a much larger surface through which to lose heat, but the gain comes from another source and is sufficient to entirely counterbalance the effect of a larger cylinder surface.

Less Condensation. When very high pressure steam and a large ratio of expansion is used, the difference between the temperature of the entering and of the exhaust steam is great. For instance, suppose steam at 160 pounds (gauge) pressure enters the cylinders and the exhaust pressure is 2 pounds (gauge), the difference in temperature as taken from steam tables is 370.5°−218.1°, or 152.4°. This difference becomes nearly 230 degrees if the steam is condensed to about three pounds absolute pressure. The cylinder and ports of the engine are cooled to the low temperature of the exhaust steam and, as we have seen, a considerable quantity of the entering steam is condensed to give up heat enough to raise the temperature of the cylinder to that of the entering steam. As the ratio of expansion increases, the difference in temperature increases, and consequently the amount of steam thus condensed also increases. To keep this initial condensation as small as possible, the range of temperature must be limited, that is, it must not have as great a difference between admission and exhaust. To do this the expansion of the steam must be divided between two or more cylinders.

It will be remembered that the great trouble Watt found with Newcomen's engine was its great amount of condensation, and he stated as the law which all engines should try to approach, that *the cylinder should be kept as hot as the steam which enters it.* This is to avoid condensation when steam first enters. If, instead of expand-

ing the steam in one cylinder, it be expanded partly in one and then finished in another, it will have passed out of the first cylinder before its temperature has dropped a great deal, and consequently the cylinder walls will be hotter than they would have been if the expansion had taken place entirely in one cylinder. This would then reduce the amount of steam condensed. The importance of this may not be evident at first, but it makes a great difference in the economy of the engine. If there is less condensation, there will be less moisture to re-evaporate, and consequently less exhaust waste, hence there will be a saving in two ways.

Methods of Compounding. In a compound engine the steam is first admitted to the smaller, or high pressure, cylinder and then exhausted into the larger, or low pressure, cylinder.

Suppose steam at 160 pounds (gauge) pressure is admitted to a cylinder, and the ratio of expansion is such that the steam is exhausted at about 60 pounds (gauge) pressure; then the difference of temperature is 370.5°−307°, or 63.5°.

If now the steam when exhausted from the first cylinder enters a second and is allowed to complete its expansion, so that the exhaust pressure is about two pounds (gauge) pressure, the difference of temperature in the cylinder will be 307°−218.5°, or 88.5°.

Then for the simple engine, if the exhaust pressure is two pounds (gauge), the difference of temperature is 152 degrees, while in the compound engine this difference is divided into two parts, 63.5 degrees and 88.5 degrees. The cylinder condensation for both cylinders of the compound engine will be much less than if the total expansion took place in a single cylinder. The cylinders should be so proportioned that the same quantity of work may be done in each.

If there are two stages of expansion, the engine is called simply *compound;* three stages, *triple;* and four, *quadruple.*

Exhaust Waste Utilized. Besides reducing the excessive condensation there is still another gain in using multiple expansion. It has been shown how much heat is lost by the exhaust waste, which in the simple engine blows into the air or into the condenser and is entirely lost. In the multiple expansion engine the exhaust and re-evaporation from one cylinder passes into the next and does work there; furthermore, any leakage from the high pressure cylinder is also allowed to do work in the low.

JACKETING

The most primitive method of effecting steam economy is by jacketing, which principle Watt early recognized and adopted. This method reduces the loss due to cylinder condensation by supplying heat to the steam while it is in the cylinder, that is, by surrounding the cylinder with an iron casting and allowing live steam to circulate in the annular space thus formed. The cylinder covers are also made hollow to permit a circulation of live steam. A cylinder having the annular space (*A*, Fig. 45) filled with steam is said to be jacketed. A lining *L* is often used in jacketed cylinders.

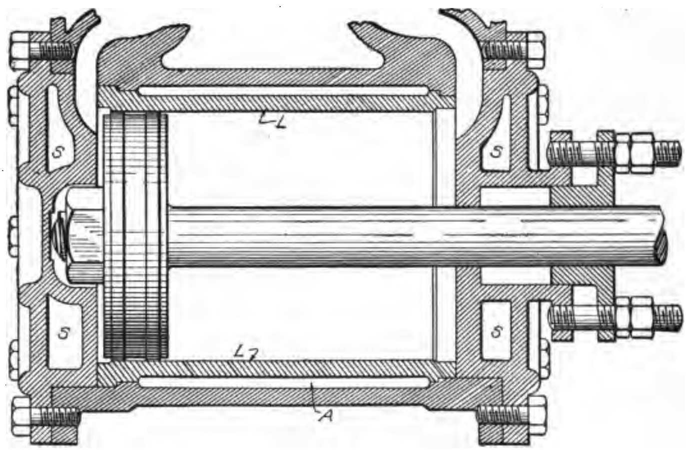


Fig. 45. Section of Steam Engine Cylinder Showing Method of Jacketing

Function of Jacket. The function of the jacket is to supply heat to the cylinder walls to make up for that abstracted during expansion and exhaust, so that at admission the cylinder will be as hot as possible. The result is, that the difference in temperature between the cylinder walls and the entering steam is considerably less than in engines where no jacket is used. Condensation is therefore reduced and, since heat flows from the jacket to the cylinder during expansion, a much larger amount of this condensation is re-evaporated before release and it thus has an opportunity to do some work in the cylinder. This leaves a comparatively small amount of exhaust waste and the heat thus abstracted is made up from the steam in the jacket. Since a large amount of heat is given

up by the jacket steam, a good deal of it must be condensed. Thus the question is asked: "What is the advantage of this method over that of allowing the entering steam to supply the heat by its own condensation?" This question is answered briefly as follows:

The loss of heat by condensing the steam would be less if the inside of the cylinder could be kept dry. It has been indicated how the moisture that collects by condensation is re-evaporated during expansion and exhaust because the pressure falls and the cylinder walls are hotter than the steam. This re-evaporation takes place at the expense of the heat in the cylinder walls and they are thus cooled. It has already been shown that a great many British Thermal Units are thus taken from the cylinder and thrown out at exhaust at every stroke. Now if the inside of the cylinder can be kept dry so that there will be little or no re-evaporation at exhaust, it will cause a considerable saving. The steam that condenses in the jacket does not re-evaporate in it; but is returned to the boiler as feed water, so that the only heat lost is the latent heat given up during condensation. If the cylinder is heated from within, both the latent heat given up by condensation and the latent heat required for re-evaporation are lost.

In a triple expansion engine there is one distinct advantage in allowing condensation in the cylinder, for this moisture acts as a lubricant, and as the heat of re-evaporation passes into the next cylinder and there does work, there is very little loss.

Saving Due to Jacketing. It is evident that a large part of the heat of the steam jacket flows to the cylinder during exhaust and is thus entirely lost in the simple engine. In the triple engine, however, this heat passes into the intermediate and low pressure cylinders; consequently we might expect a greater gain from using a jacket on a triple engine than on a large simple engine. The main advantage of the jacket has been previously pointed out, and as in all cases the gain is small, there is to be found a considerable diversity of opinion as to its real advantages. On some engines there is undoubtedly little if any gain, the largest gain being in the smaller engines of, say, 200 horsepower and under. On very small engines such as a 5-inch × 10-inch engine when developing only one and one-half horsepower under light load, the gain is as much as 30 per cent. On a 10 horsepower engine the gain might be as

much as 25 per cent, while on engines of about 200 horsepower the gain would probably be 5 to 10 per cent for simple condensing and compound condensing, and from 10 to 15 per cent for triple expansion. The saving on large engines of, say, 1,000 horsepower, is very small, the reason being that large engines offer less cylinder surface per unit of volume than small ones, and hence have proportionately less cylinder condensation. The very small engines, in which the gain would be the greatest, are seldom jacketed, because they are built for inexpensive machines and the first cost is of more consequence than the economy of operation. Owing to the cost of construction and the care necessary to keep jackets operative, the use of the jacket has gradually diminished. Furthermore, the introduction of the high speed and compound engines, as well as the use of superheated steam, has reduced the advantage of jacketing to relative insignificance.

SUPERHEATING

General Practice. The use of superheated steam is rather a modern practice, although for many years previous to its adoption engineers had appreciated its value in producing steam engine economy. The reason for its delayed adoption in a practical way was due to the mechanical difficulties met with in superheating the steam and also to the increased cost of maintenance produced by its use. In recent years both of the objectionable features above mentioned have been, in a large measure, overcome, so that today superheat is being used in a large number of power plants, and also in steam locomotives.

Before describing a superheater it may perhaps be well to clearly define what is meant by superheated steam. Water, when confined in a vessel and heated sufficiently, turns into steam, which, if some water still remains, is spoken of as saturated steam. Saturated steam when further heated becomes superheated steam, if it is separated from the water. To bring about this separation, a superheater is necessary. Superheaters vary considerably in details of construction according to the service for which they are designed, there being, for instance, quite a difference between the superheater designed for a stationary plant and one designed for a locomotive.

Foster Superheater. A Foster superheater as applied to a water tube boiler is illustrated in Fig. 46. The superheating element

is shown as *B*, which is connected to the steam space of the boiler by the pipe *A*. The saturated steam from the boiler passes through the pipe *A*, through the superheater *B*, and then is conveyed to the engine through the valve *C*. In this installation the superheater is placed in the passage provided for the transmission gases to the chimney, hence it is heated by what would otherwise be lost heat. The manner of installing superheaters varies a great deal. Some are entirely separated from the boiler, being self-contained and supplied with a grate for separate firing.

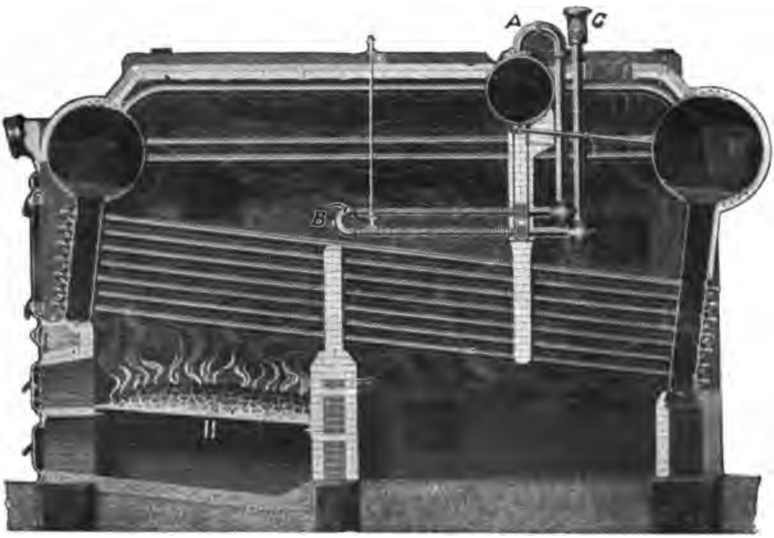


Fig. 46. Section of Water-Tube Boiler Showing Application of Foster Superheater

The Foster superheater, Figs. 46 and 47, is made up of a number of elements placed parallel to each other, each of which consists of two straight steel tubes, one inside of the other. The elements are joined at one end to manifolds or connecting headers, and at the other end to return headers for which return bends are often substituted. On the outside of the tubes *B*, Fig. 47, are fitted a series of cast-iron annular flanges *D*, placed close to each other and carefully fitted to the tube so as to be practically integral with it, at the same time exposing an external surface of cast iron, which metal is best adapted to resist the action of the heated gases. The rings are carefully bored to gauge and shrunk on the tubes. Once being in posi-

tion, the rings and tubes act virtually as a unit. As the coefficient of expansion of steel is a trifle greater than that of cast iron, the rings grip the tubes even tighter when in service. This form of construction is flexible and durable. It provides a section of great strength and entire freedom from internal strains. The mass of metal in the tubes and covering acts as a reservoir for heat, which is imparted to the steam evenly, tending to secure a constant temperature of steam, even though the temperature of the hot gases does fluctuate. The seamless drawn tube secures great initial strength, which is reinforced by the rings shrunk on the outside. Inside of the elements there are placed other tubes C of wrought iron, which are centrally supported by means of knobs or buttons regularly spaced throughout their length. These inner tubes are closed at the ends. A thin annular passage E from the steam is thus formed between the inner and the outer tubes. The steam clinging closely to the heating surface is quickly heated in the most efficient manner.

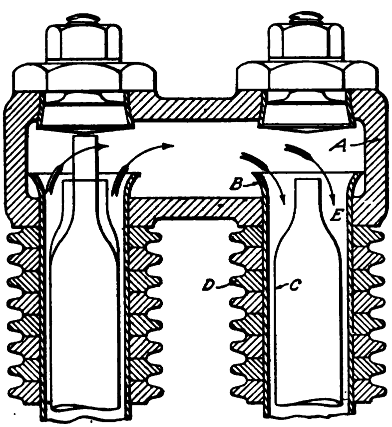


Fig. 47. Section of Foster Superheater Tubes

The superheater must be as free as possible from the liability of burning out in case of a chance overheating of the exposed surfaces. The circulation must be properly distributed throughout the superheater at full load as well as at partial loads. The various parts must be accessible for inspection, both externally and internally, and must be readily renewable or easily repaired. There must be provision for free expansion and contraction of the various parts. The supporting arrangement must be carefully worked out. Cast iron has given excellent results in producing durable superheaters and has been extensively used because of its ability to withstand high temperatures. For high steam pressures, however, cast iron is not considered safe, and has given way to the use of seamless steel tubes, which are homogeneous and strong, but lack the heat-resisting qualities of cast iron. It is evident that a combination of these two metals will preserve the good qualities of both.

Separately-Fired Superheater. A type of superheater differing radically from the one previously described is illustrated in Fig. 48. It is a separately-fired superheater and its construction is very similar to the Stirling water-tube boiler. The saturated steam from the main boiler plant enters the rear superheater drum *1*, passes through the rear bank of tubes *7* into the lower drum *2*, thence to the upper drum *3*, from which it passes into the pipe line through the

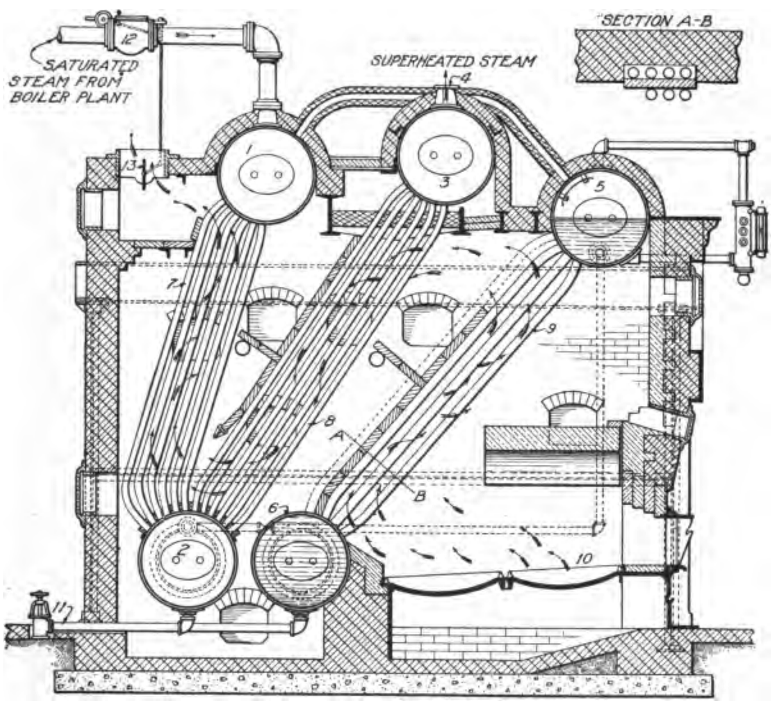


Fig. 48. Section of a Separately-Fired Type of Superheater

opening *4*. The furnace is similar to that used in the standard design of Stirling boiler. To protect the superheater tubes from high temperatures of the furnace, a sufficient amount of boiler heating surface, as drums *5* and *6* and bank of tubes *9*, is located in front of the superheater proper in order to reduce the temperature of the gases to about 1,500 degrees by the time they reach the superheater. The builders state that when the gas temperature reaches 1,500 degrees in the standard boiler, 19 per cent of the boiler heating sur-

face has been swept over by the gases, 50 per cent of the steam produced by the boiler has been generated, and the boiler heating surface per horsepower is 3.8 square feet. Consequently in the independently fired superheater shown in Fig. 48, 50 per cent of the heat absorbed is used to generate the steam which is added to steam furnished by the main boiler plant and hence increases the capacity of the plant in proportion. The remaining 50 per cent of the heat, a portion of which passes out the stack, is absorbed by the superheater, and superheats both the steam from the main boiler plant and that from the front bank of water tubes. The superheater, because of the front generator set, will produce about 12 per cent of the amount of steam furnished by the main boiler plant. As a further precaution against any possible overheating of the superheater tubes near the furnace, a flap valve *12* is placed in the pipe conveying saturated steam to the superheater, as shown in Fig. 48. The spindle of this valve is connected by links to the superheater damper *13*, so that the damper to the opening is regulated according to the quantity of steam flowing into the superheater. If the steam flow stops, the valve *12* drops to its seat and the damper *13* is closed. Independently fired superheaters are furnished in any desired capacity, suitable for any degree of superheat up to about 300° F. The upper water drum *5* and the lower superheater drum *2* are connected by piping, hence, if desired, the superheater sections may be flooded, converting the whole into a saturated steam boiler.

Purposes of Superheaters. These two types of superheaters illustrated and described will suffice, and we may now direct our attention to a study of the purposes of the superheater and to some consideration of the economy secured by its use. The purposes of superheating steam, as practiced in the past and as recognized at present, are, according to Thurston, the following:

(1) Raising the temperature which constitutes the upper limit in the operation of the heat-engine in such a manner as to increase the thermodynamic efficiency of the working fluid.

(2) To so surcharge the steam with heat that it may surrender as much as may be required to prevent initial condensation at entrance into cylinder and still perform the work of expansion without condensation or serious cooling of the surrounding walls of the cylinder.

(3) To make the weight of the steam entering the condenser and its final heat charge a minimum, with a view to the reduction of the volume of the condensing water and magnitude and cost of the air pump and condenser system to a minimum.

(4) To reduce the back pressure and thus to increase the power developed from a given charge of steam and efficiency of the engine.

(5) To increase the efficiency of the boilers both by the reduction of the quantity of the steam demanded from the original heating surface and by increasing the area of the heating surface employed to absorb the heat of the furnace, and flue gases, and also by evading the waste consequent upon the production of wet steam.

If the steam entering a cylinder is only superheated enough to give dry saturated steam at cut-off, the range of temperature $\frac{T_1 - T_2}{T_1}$ of the Carnot cycle is interchanged and there is, therefore, no increase of economy from item 1. The other four sources of economy depend upon one fundamental fact—the poor conductivity of dry steam. To the property of non-conductivity of heat of superheated steam, is due its great advantage. On entering a cool cylinder it slowly gives up its heat, and if the degree of superheat is sufficient there will be little or no initial condensation. The degree to which steam should be superheated is still a debated point, some engineers contending that only a very moderate degree of superheat of about 100 degrees is sufficient, whereas others maintain that no real economy is obtained with less than 200 degrees or over. When a high degree of superheat was first used, difficulties were encountered such as the disintegration of the valves, valve seats, packing rings, and other parts subjected to the action of the superheated steam. Lubrication was also interfered with, since many of the oils used were not suited for such high temperatures. All of these difficulties no doubt account for the one time wide-spread objection to high degrees of superheat, but in recent years they have in a large measure been overcome. The author is familiar with the performance of a simple slide valve locomotive which has been in operation for several years under degrees of superheat ranging from 80 degrees to 214 degrees, during which time no trouble has been experienced with the valves or with the lubrication. Many European locomotives have been satisfactorily operated with high degrees of superheat, which insures the passage of steam through the cylinder with but little or no condensation.

Economic Advantages. The economy obtained by the use of superheat has been clearly demonstrated by a large number of practical tests both upon stationary engines and upon locomotives. It is to be noted also, that about the same per cent of economy has been obtained on the various types of engines tested, the stationary tests corroborating the results obtained upon the locomotive and *vice versa.* The various tests indicate a saving of from 12 to 15 per cent of the amount of steam used by the engine per indicated horsepower per hour, and a saving of coal from 20 to 25 per cent. Another very significant thing that has been determined is that the output of power has been increased from 20 to 30 per cent, depending upon the conditions. These three items of saving have hastened the installment of a large number of superheaters, so that at the present time thousands of locomotives in Europe are equipped with superheaters, and in the United States and Canada over 1,500 locomotives are so equipped. It seems that the railroads have been quicker to take up the idea of installing superheaters than have other industries, so that not nearly so many superheaters are found in stationary service.

It is to be noted that the greatest gains from the use of superheaters are to be expected in the more uneconomical plants. That is, the per cent of saving by the use of superheated steam in a simple engine would be greater than for a compound engine, and for a compound engine as compared with a triple expansion engine. Several prominent engineers have advised the reduction of steam pressures with a relative increase in diameter of cylinders and the use of superheated steam. The combination of a simple engine with low steam pressure and superheated steam will give an increased output of power at a small cost, a result desired by all operators.

CONDENSERS

When low pressure steam is cooled it gives up its latent heat, that is, it changes from a vapor to a liquid, and, as a liquid occupies much less space than an equal weight of its vapor, the changing of the steam to water greatly reduces the pressure. Therefore, by cooling the steam in an engine cylinder in front of the piston, the back pressure, or resistance, is reduced, which, in turn, reduces the pressure necessary to push the piston through the stroke and, there-

fore, lessens the steam required to do the work. This cooling is accomplished by some form of condenser.

Theory of Condenser Action. *Back Pressure.* In the ordinary non-condensing engine, steam can not be exhausted below a pressure of 14.7 pounds absolute, because the atmosphere exerts that amount of pressure at the opening of the exhaust pipe. In fact, this 14.7 pounds is the theoretical limit only, and in practice the exhaust is always a little above this because of resistance in the exhaust ports and exhaust pipe; so that 17 or even 18 pounds absolute back pressure is more nearly the conditions of actual service.

During the forward stroke, steam expands from the pressure at admission to a much lower pressure at release; then the valve opens for the return stroke giving full steam pressure on one side of the piston and the pressure of exhaust on the other side, the latter acting against the piston and against the force of the incoming steam. If all of this back pressure could be removed so that there would be a vacuum on the exhaust side of the piston, the power of the engine would be increased by just so many pounds of mean efficiency pressure, and in addition to this the steam could expand to a very much lower pressure and therefore work with greater economy.

Effect of Condensation. One pound of steam at 17 pounds absolute pressure occupies 23.22 cubic feet of space in the cylinder of the engine, but one pound of water in the condenser occupies only about 0.016 cubic feet, which makes the steam occupy nearly 1,450 times as much space as the water into which it condenses. If then, the exhaust steam could be condensed instantly, the back pressure would be reduced almost to zero and the engine would exhaust into a vacuum.

Unfortunately the mere condensation of the steam will not give a perfect vacuum because of the air, always present in the water, which comes over from the boiler. Moreover, the condensed water is hot, and the vapor rising from it in the condensing chamber, together with the air and some leakage would spoil the vacuum were it not for the air pump, which removes the air and condensed steam. Even with the best air pump it would be impossible to maintain a perfect vacuum, but a vacuum of 26 inches, which corresponds to about 2 pounds absolute pressure, can readily be maintained in good practice.

It is well known that a certain amount of heat is required to change one pound of water at a given temperature into steam at the same temperature; this is called the latent heat of vaporization. If the steam condenses, it must give up this latent heat. The easiest ways of doing this are either to let the steam come in contact with pipes through which cold water is circulated, as in a surface condenser, or mingle with a spray of water, as in a jet condenser. These two types will now be discussed.

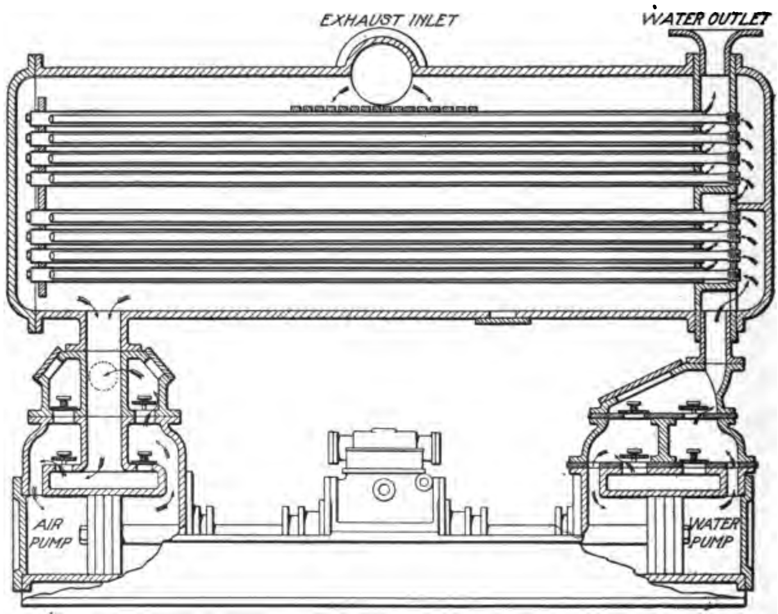


Fig. 49. Section of Steam Condenser of the Surface Type

Types of Condensers. Condensers may be divided into two general classes as follows:

(1) Surface condensers in which the cooling water is separated from the steam, usually by metallic surfaces in the form of tubes, the cooling water circulating on one side of this surface and the steam coming in contact with the metal on the other side.

(2) Jet condensers, including barometric condensers, siphon, condensers, ejector condensers, etc., in which the cooling water mingles with the steam to be condensed.

Surface Type. The condenser shown in section in Fig. 49 is one form of the surface type, in which the air pump and the circulating

pump are both direct acting and both operated by the same steam cylinder. The cool condensing water is drawn from the supply into the circulating or water pump and is forced up through the valves and water inlet to the condenser. It flows, as indicated by the arrows, through the inner tubes of the lower section, then back through the space between the inner and the outer tubes. The water then passes upward and through the upper section, as it did in the lower, then it passes out of the condenser through the water outlet, taking with it the heat it has received from the steam.

The exhaust steam from the engine enters at the exhaust inlet and comes in contact with the perforated plate, which causes it to spread. The steam expanding in the condenser comes in contact

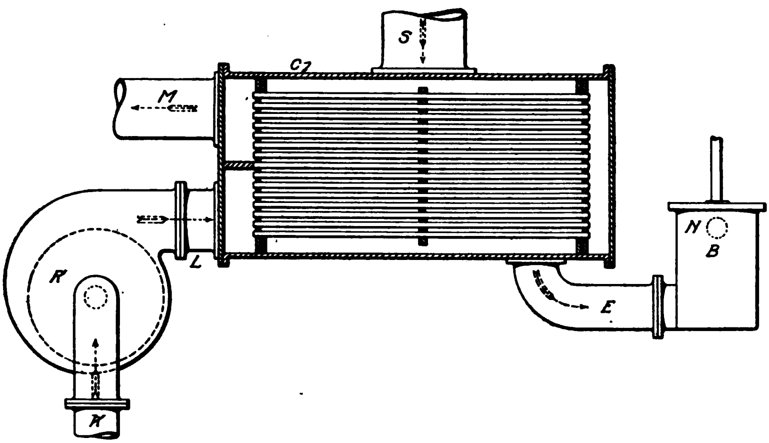


Fig. 50. Diagram Showing Relation of Surface Condenser to the Pumps Necessary for Proper Operation

with the tubes, through which cool water is circulating, and condenses. The air pump draws the air and condensed steam out of the condenser and thus maintains a partial vacuum. This causes the exhaust steam in the engine cylinder to be drawn into the condenser, at the bottom of which it collects as it condenses and is drawn into the air pump cylinder and discharged while heated to the hot well of the boiler. The use of this hot water as feed water effects a considerable saving, but the great advantage of the condenser is the reduction of the back pressure.

Hot water can not be used by an ordinary pump as easily as cold water because of the pressure of the vapor which arises from

the hot water. In the condenser shown, the water and air pumps are run by the piston in the steam cylinder. Sometimes these pumps are connected to the main engine and receive motion from the shaft or crosshead.

The general arrangement of the surface condenser with the necessary pumps is shown in Fig. 50. The cooling water enters through the pipe *K* and flows to the circulating pump *R*, which forces the water into the condenser through the pipe *L*. In case the water enters the condenser under pressure from city mains, no circulating pump is necessary. After flowing through the tubes it leaves the condenser by means of the exit *M* and flows away. Exhaust steam enters at *S* and is condensed by coming in contact with the cold tubes; the water (condensed steam) then falls to the bottom of the condenser and flows to the air pump *B* by the pipe *E*. The air pump removes the air, vapor, and condensed steam from the

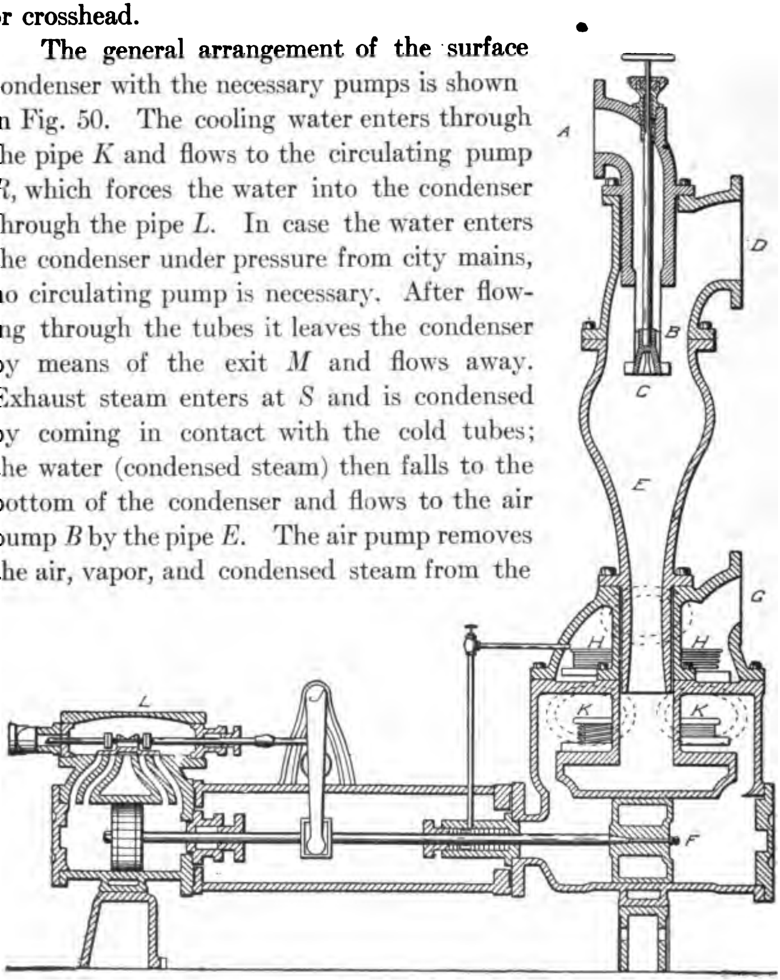


Fig. 51. Longitudinal Section of Independent Jet Condenser and Pump

condenser and forces it through the pipe *N* into the hot well from which it goes to the boilers or to the feed tank.

Circulating Pump. The circulating pump, when separate from the condenser, is usually of the centrifugal type. This pump consists of a fan or wheel which is made up of a central web (or hub) and

arms (or vanes). The vanes are curved and as the water is drawn in at the central part, the vanes throw it off at the circumference. A suitable casing directs the flow. This type of pump is advantageous because there are no valves to get out of order and, as the lift is little, if any, the pump will discharge a large volume of water in a nearly constant stream. The circulating pump is usually so placed that the water flows to it under a slight head. The pump is driven by an independent engine so that the circulating water may cool the condenser even if the main engine is not working.

Jet Type. Fig. 51 illustrates the longitudinal section of an independent jet condenser and pump. The cold water used to condense the steam enters at *A*, passes down the spray pipe *B*, and is broken into a fine spray by means of the spray cone *C*. This action insures a rapid and thorough mixing of the steam and water and consequently a rapid condensation. The exhaust steam enters at *D* with a comparatively high velocity, which is imparted to the water. The whole mixture of water, steam, and vapor passes at high velocity through the conical chamber *E* to the pump cylinder *F*, where it is forced into the pipe *G*. The spray cone is adjusted by means of the stem which passes through the stuffing box at the top of the condenser. The valves are shown at *H* and *K*. The steam end of the pump is at *L*.

In Fig. 52 a jet condenser is shown connected to a stationary engine. The exhaust pipe leads from the engine to the condenser, the arrows indicating the direction of the flow. Cold water enters the condenser through the pipe shown. Part of the mixture of exhaust steam and condensed water goes to the feed-water heater, which is kept nearly full; the rest passes to the sewer. The heater is placed a little above the feed pump, so that the water will enter the pump under a slight head, because the pump can not raise water warmed by exhaust steam as readily as cold water.

Relative Merits of Jet and Surface Condensers. In the jet condenser the steam, as soon as condensed, becomes mixed with the cooling water, and if the latter should be unsuitable for boiler-feed because of scale-forming impurities, acids, salts, etc., the pure distilled water represented by the condensed steam is wasted, and if it were necessary to purchase other water for boiler-feeding, this might represent a considerable waste of money. On the other hand, if the

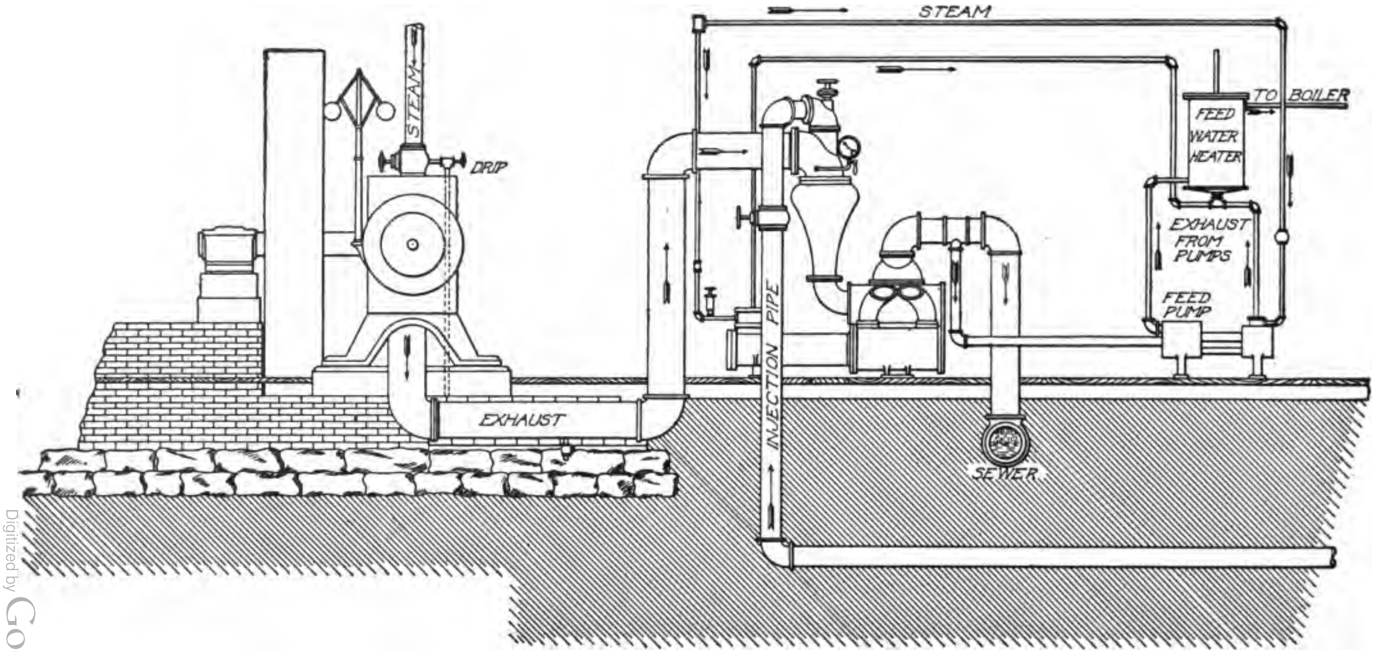


Fig. 52. Diagram Showing Stationary Engine with Connection to Jet Condenser, and Other Necessary Appliances

cooling water is suitable for boiler-feeding or if a fresh supply of good water is easily obtainable, the jet condenser, because of its simplicity and low cost, is unexcelled. Surface condensers are recommended where the cooling water is unfitted for boiler-feed and where no suitable and cheap supply of pure boiler-feed water is available. Condensed steam from a surface condenser makes the best boiler-feed water, being in fact pure distilled water entirely free from scale-forming matter and containing a considerable amount of heat, as compared with cold feed water. If the exhaust from reciprocating engines is to be condensed and used as boiler-feed water, a suitable oil separator should be interposed in the exhaust pipe between the engine and the condenser. Another advantage of the surface condenser as compared with the jet condenser is that there is no danger, in case of failure of vacuum pumps, of the circulating water backing up into the engine cylinder and wrecking the engine.

Effect of Condenser on Efficiency. It has already been stated heat there is a gain in thermal efficiency by running an engine condensing, but it will be more clearly seen by considering a few figures. The thermal efficiency may be expressed by the previously mentioned formula

$$E=\frac{T_1-T_2}{T_1}$$

This efficiency may be increased if T_1 can be made larger—which would happen if the boiler pressure were increased—or if T_2 can be made smaller, which would result from reducing the back pressure by condensing. If the boiler pressure is raised, both the numerator and denominator of the fraction will increase, and the value of the fraction will be but slightly greater. If, however, the back pressure is reduced, the numerator T_1-T_2 will be larger, while the denominator T_1 will remain the same. It is apparent that this will cause a much greater increase in efficiency than raising the boiler pressure a like amount.

Suppose an engine is supplied with steam at 85.3 pounds (gauge) pressure and it exhausts at 3.3 pounds (gauge) pressure. The absolute temperature corresponding to 85.3+14.7, or 100 pounds pressure, is 327.58+461, or 788.58 degrees, and the absolute temperature corresponding to 3.3+14.7, or 18 pounds pressure, is 222.40+461, or

TABLE

Increase in Efficiency by Use of Condenser for Various Engines

Type of Engine	Feed Water per Indicated Horsepower				Per Cent Gained by Condenser
	Non-Condensing		Condensing		
	Probable Limits Pounds	Assumed for Comparison Pounds	Probable Limits Pounds	Assumed for Comparison Pounds	
Simple High Speed	35 to 26	33	25 to 19	22	33
Simple Low Speed	32 to 24	29	24 to 18	20	31
Compound High Speed	30 to 22	26	24 to 16	20	23
Compound Low Speed	— —	24	20 to 12¾	18	25
Triple Exp. High Speed	27 to 21	24	23 to 14	17	29
Triple Exp. Low Speed	— —	—	18 to 12	—	—

683.40 degrees. Then the thermal efficiency determined from the formula becomes

$$E=\frac{T_1-T_2}{T_1}=\frac{788.58-683.40}{788.58}$$
$$=.133, \text{ or } 13.3 \text{ per cent}$$

If the boiler pressure were raised to 140 pounds absolute the efficiency would be

$$E=\frac{813.85-683.40}{813.85}$$
$$=.16, \text{ or } 16 \text{ per cent}$$

If instead of increasing the boiler pressure a condenser is used and the exhaust pressure reduced to 4 pounds (absolute), the efficiency becomes

$$E=\frac{788.58-614.09}{788.58}=.221, \text{ or } 22.1 \text{ per cent}$$

Thus it is seen that if the exhaust pressure is lowered 14 pounds absolute there will be a greater increase in efficiency than if the boiler pressure is raised 40 pounds.

The per cent of efficiency that is obtained by the use of a condenser is shown in Table I.

Cost of Cooling Water Determines Condenser Economy. While the above figures are very encouraging, yet conditions may arise where the per cent of gain may be materially lessened or entirely lost,

due to the cost of water. Condensing engines require from 20 to 30 pounds of cooling water to condense each pound of steam used, depending on the necessary temperature. Thus it can be seen that the quantity of cooling water is relatively very large, and if it is purchased from a water company, quite an item is added to the yearly expense account for the one item of water. If, however, some means could be provided whereby the circulating water as it issues from the condenser could be cooled and then used over again in the condenser, the non-condensing engine could be run condensing, thus taking advantage of all the benefits due to the use of reduced back pressure and heating of the feed water. This has been attempted by conducting the heated discharge water to a pond, where it is allowed to cool to a lower temperature before being used again. Another plan is to place in the yard or on the roof of the building large shallow pans, in which the water is cooled by being exposed to the atmosphere. These methods are unsatisfactory on account of the considerable area necessary and the slow action. In addition, they are uncertain, because they are dependent upon atmospheric conditions.

Cooling Tower and Water Table. A more efficient and at the same time more expensive process is to use a cooling tower or a water table. Fig. 53 illustrates the general arrangement of a cooling tower located upon the roof of a building. The discharge from the condenser is led, as shown by the arrows, to the top of the cooling tower, where it is cooled before being returned to the condenser. This cooling is effected by distributing the water, by a system of piping, to the upper edge of a series of mats or slats, over the surface of which the water flows in a thin film to a reservoir which is situated in the bottom of the cooling tower. The mats partially interrupt the flow and, by breaking up the water in small streams, cause new portions to be exposed to the cooling effect of the air currents. The water from the reservoir then flows downward through the suction pipe and is pumped by the circulating pump through the condenser. After passing through the condenser and absorbing heat from the exhaust steam, it rises through the discharge pipe and commences the circuit over again.

The tower may have several arrangements and be made of various materials. A satisfactory form is constructed of steel plates

within the tower, or a large number of mats of steel wire cloth galvanized after weaving. The tower may be supported upon a proper foundation or upon legs, instead of being situated on the top of a building, as the one shown in the illustration.

To assist in the cooling of the water, the air is often made to circulate rapidly by means of a fan, which forces the air into the

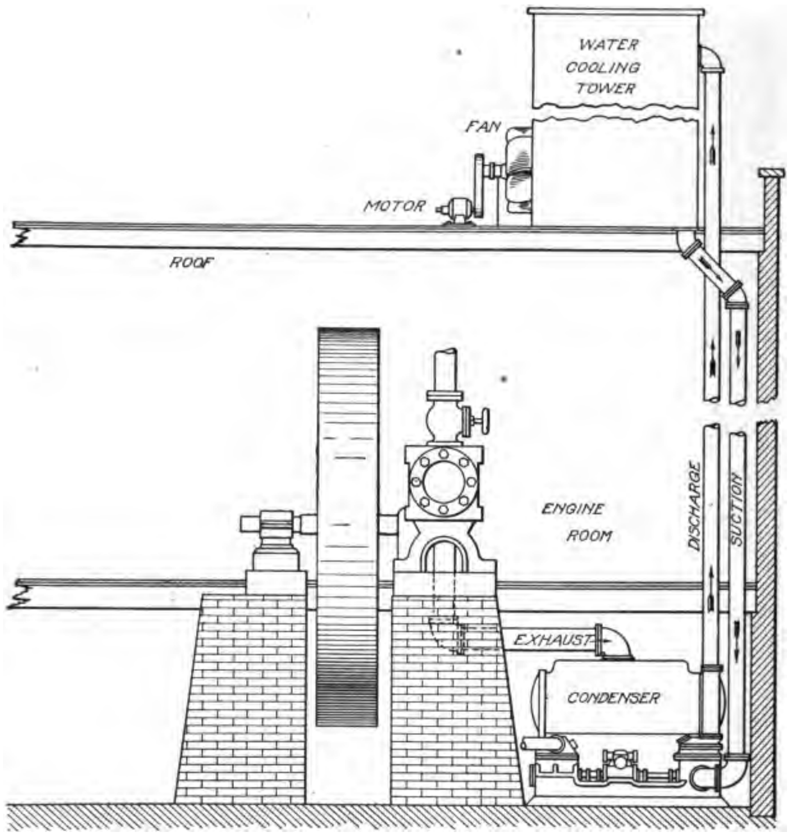


Fig. 53. Diagram of Stationary Engine with Connections to Water Cooling Tower on Roof of Building

lower part of the tower and upward through the mats. This fan may be driven by an electric motor, by a line of shafting, or by a small independent engine.

In case the fan is not used, the mats are arranged so that they are exposed to the atmosphere. This of course necessitates the removal of the steel casing. Usually the fanless tower must be

placed at the top of a high building or in some position where the currents of air can readily circulate through the mats.

With an efficient type of cooling tower, the water may be reduced from 30 to 50 degrees, thus allowing a vacuum of from 22 to 26 inches. This will, of course, greatly increase the economy of the plant and allow the heated feed water to be returned to the boiler.

The water table is usually made of wooden slats placed in the ground near the plant. After trickling over the slats and becoming cooled by the air, it collects in the bottom of the reservoir and is then pumped into the condenser.

Amount of Cooling Water Per Pound of Steam. Besides condensing the steam, the injection water cools it still further, so that more than merely the latent heat is removed from it. If exhaust steam enters the condenser at a temperature t_1, it contains a certain amount of heat, known as *total heat at temperature* t_1. If it is condensed and cooled to a temperature t_2, at which it leaves the condenser, it then contains a certain amount of heat, known as *total heat at temperature* t_2.

If A represents the total heat at t_1 and B represents the heat of the liquid at t_2, then the heat given up by one pound of condensed steam is equal to $(A-B)$ British Thermal Units, provided the exhaust that enters the condenser is dry saturated steam. If C is the temperature of the injection or cooling water and D is the temperature of the discharge water, then every pound of cooling water absorbs approximately one British Thermal Unit for every degree rise in the temperature, or we may say that the heat absorbed is equal to $(D-C)$ British Thermal Units per pound of cooling water. Then it will take as many pounds of water W to absorb $(A-B)$ heat units as $(D-C)$ is contained in $(A-B)$. This may be expressed thus

$$W=\frac{(A-B)}{(D-C)}$$

Therefore, W represents the number of pounds of water required per pound of steam condensed.

EXAMPLE 1. Suppose steam is expanded in an engine to 4 pounds absolute pressure. If the initial temperature of the cooling water is 45 degrees, and the condenser is of the surface type, discharging water at 120 degrees,

and the temperature of the condensed steam is 130 degrees, how many pounds of cooling water are required per pound of steam?

SOLUTION. By consulting the steam tables we find the total heat of steam at 4 pounds pressure to be 1,128.6 British Thermal Units. The heat of the liquid in the condensed steam at 130 degrees is 98.1 British Thermal Units. Then

$$W = \frac{1128.6 - 98.1}{120 - 45}$$
$$= 13.74 \text{ pounds}$$

EXAMPLE 2. Suppose steam at 6 pounds absolute pressure exhausts into a jet condenser. The temperature of the injection water is 50 degrees and the discharge is 120 degrees. How many pounds of water are necessary to condense 8 pounds of steam?

SOLUTION. In the jet condenser the temperature of the condensed steam and the discharge water is the same. We find from the steam tables that the total heat of steam at 6 pounds absolute is 1,133.8 British Thermal Units, and the heat of the liquid in the condensed steam at 120 degrees is 88.1 British Thermal Units. Then as before

$$W = \frac{1133.8 - 88.1}{120 - 50}$$
$$= 14.94$$

Therefore, 8 pounds of water will require 14.94×8, or 119.52 pounds.

The above calculation can not be relied upon to any great extent for we seldom know the true condition in the condenser, and it would be of little value to us if we did know, as the exact condition will change considerably. In practice it is customary to allow for about twice as much water as the above calculation would require. These figures give us a fair idea of the necessary sizes of the pipes and passages leading to the condenser, and give a basis for estimating the dimensions of the air pump.

Cooling Surface in Surface Condensers. The amount of surface required to condense the steam in surface condensers depends upon the conductivity of the metal, the condition of the tubes and their thickness, and the difference in temperature between the two sides. The tubes of a condenser are much thinner than boiler tubes, hence we might expect them to be more efficient in condensing the steam than the boiler tubes are in evaporating water. It has been found in actual practice, that a surface condenser receiving cooling water at 60 degrees and discharging it at 120 degrees will condense from 10 to 20 pounds of steam per square foot of the tube surface per hour. An average of 13 pounds per square foot of surface per hour

is considered a fair one. With exhaust pressure from 6 to 30 pounds absolute, it has been found that an allowance of 1.5 to 3.0 square feet of cooling surface per indicated horsepower is sufficient, when the initial temperature of cooling water is 60 degrees and the final temperature is 120 degrees.

It is evident that the amount of surface will depend upon the quantity of steam used per hour by the engine, the pressure and temperature of the exhaust, and the temperature of the cooling water and discharge. There must also be an allowance for inefficient work after the condenser has become fouled with service. All these conditions make the problem so uncertain that calculations by means of formulas are likely to be untrustworthy, and it is best at all times to make estimates from the figures given for similar conditions in actual service.

Feed Water Heaters. In many places where water is expensive and the condensing engines can not be run economically, a very considerable saving can be effected either by allowing the exhaust steam to condense into a feed water heater, thus saving the heat that would otherwise be wasted, or by using the exhaust steam for heating purposes. Of course in such cases the steam consumption of the engine is high, but if proper allowance is made for the heat used for other purposes, the actual fuel consumption rightfully charged to the engine is not excessive. If the feed water is heated by waste gases, then the gain belongs to the boiler and not to the engine.

ANALYSIS OF ENGINE MECHANISMS

CRANK EFFORT

In the steam engine the steam exerts a pressure on the crank pin through the piston rod and connecting rod. When the crank is at the dead center, the entire pressure is on the bearing of the crank shaft, and there is no tendency to turn the crank. As the crank pin moves from the dead center, the tendency increases until it reaches a maximum and then decreases until, at the other dead center, it is zero again. If the connecting rod were of infinite length and steam were admitted throughout the whole stroke, the maximum tendency, or the maximum turning moment as it is called, would occur with the crank at right angles to the line connecting the dead points.

Variable Thrust. In the actual engine the thrust along the rod is constantly varying even though the pressure on the piston remains the same. This is due to the angularity of the connecting rod. The turning moment is always equal to the thrust along the connecting rod multiplied by the perpendicular distance from the connecting rod to the center of the shaft. If the steam pressure on the piston remains constant, the maximum turning moment occurs when the connecting rod is at right angles to the crank, for in this position the perpendicular distance from the rod to the center of the shaft is a maximum and equal to the length of the crank; and, as the rod makes its greatest angle with the line connecting the dead center at this point, the thrust along it will also be a maximum. If the cut-off is very early, one-quarter stroke for instance, the maximum thrust along the rod will occur earlier than at the point previously mentioned, but the leverage of the force will be less, so that really there will be little change in the point of maximum turning moment no matter where the cut-off may occur.

Diagrams. To represent this turning moment, diagrams of crank effort may be drawn, with rectangular co-ordinates, having the crank angles represented as abscissas and the turning moments corresponding to these angles as ordinates.

FLYWHEEL

Besides the thrust of the connecting rod there must be taken into account friction and the inertia of the reciprocating parts. At first this may be thought of small consequence but with a fairly heavy piston and connecting rod it is obvious that at high speed the momentum would be great. In the case of a vertical engine, on the up stroke the steam must lift this heavy mass and impart a very considerable velocity to it, while on the down stroke the acceleration of the mass is added to the steam pressure. This makes the effective force on the up stroke less than that due to the actual steam pressure, and greater on the down stroke.

Function. In the case of a horizontal engine it is evident that while the piston can push the crank around during part of the stroke, and pull it along during another part, yet at the end of the stroke the pressure on the piston, no matter how great, can exert no turning moment on the shaft. Therefore, if some means is not pro-

vided for making the shaft turn past these points without the assistance of the piston, it may stop. This means is provided in the flywheel which is merely a heavy wheel placed on the main shaft. On account of the momentum of the flywheel it can not be stopped quickly and therefore carries the shaft around until the piston can again either push or pull.

Size of Wheel. If a long period be considered, the mean effort and the mean resistance must be equal; but during this period there are temporary changes of effort, the excesses causing increase of speed. To moderate these fluctuations several methods are employed.

The turning moment on the shaft of a single cylinder engine varies, *first*, because of the change in steam pressure, and *second*, on account of the angularity of the connecting rod. Before the piston reaches mid-stroke the turning moment is a maximum, as shown by

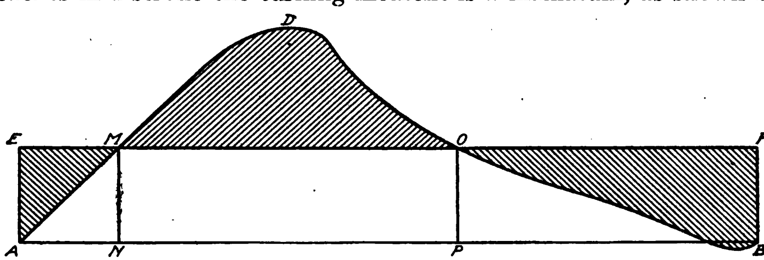


Fig. 54. Graphical Representation of Turning Moment of Crank Shaft of a Single-Cylinder Engine for One Stroke

the curve, Fig. 54. Near the ends of the stroke the turning moment diminishes and finally becomes zero. This, of course, tends to cause a corresponding change in the speed of rotation of the shaft. In order to have this speed as nearly constant as possible and to give a greater uniformity of driving power, the engine may be run at high speed. By this means the inertia of the revolving parts, such as the connecting rod and crank, causes less variation. When the work to be done is steady and always in the same direction, a heavy flywheel may be used. The heavier the flywheel, the steadier will be the motion. It is desirable, of course, in all engines to have steady motion, but in some cases it is more important than in others. For instance, in electric lighting plants it is necessary that the machinery shall move with almost perfect steadiness. It is undesirable to use larger wheels than are absolutely necessary, because of the cost of the metal, the weight on the bearings, and the danger from bursting.

Methods of Reducing Size. If the turning moment which is exerted on the shaft from the piston could be made more regular and if dead points could be avoided, it would be possible to get a steadier motion with a much smaller flywheel.

If the engine must be stopped and reversed frequently, two or more cylinders are used, being connected to the same shaft. The cranks are placed at such angles that when one is exerting its minimum rotative effort, the other is exerting its maximum, or when one is at a dead center, the other is exerting its greatest effort. These cylinders may be identically the same in dimension as is the case with most hoisting engines and with many locomotives; or the engine may be compound or triple expansion. This arrangement is also used on engines for mines, collieries, and for hoisting of any sort where ease of stopping, starting, and reversing are prerequisites. Simple expansion engines with their cranks at right angles are usually spoken of as being coupled.

The governor adjusts the power of the engine to any large variation of the resistance. The flywheel has a duty to perform which is similar to that of the governor. It is designed to adjust the effort of the engine to sudden changes of the load which may occur during a single stroke. It also equalizes the variation in rotative effort on the crank pin. The flywheel absorbs energy while the turning moment is in excess of the resistance, and restores it while the crank is at or near the dead points. During these periods the resistance is in excess of the power.

Action of Flywheel. The action of the flywheel may be represented as in Figs. 54 and 55. It will be noticed that in Fig. 54, the curve of the crank effort runs below the axis toward the end of the stroke. This is because the compression is greater than the pressure near the end of expansion, and produces a resultant pressure on the piston. In Fig. 55 the effect of compression has been neglected. Let us suppose that the resistance, or load, is uniform. In Fig. 54, the line $A\ B$ is the length of the semi-circumference of the crank pin, or the circumferential distance the crank pin moves during one stroke. The curve $A\ M\ D\ O\ B$ is the curve of turning moment for one stroke. $M\,N$ is the mean ordinate and, therefore, $A\ E\ F\ B$ represents the constant resistance. The effort and resistance must be equal if the speed is uniform; hence the area $A\ E\ F\ B$ equals

$A\,M\,D\,O\,B$. Then area $A\,E\,M$ plus area $O\,F\,B$ equals area $M\,D\,O$. At A the rotative effort is zero because the crank pin is at the dead point and from A to N, the turning moment is less than the resistance. At N the resistance and the effort are equal. From N to P the effort is in excess of the resistance. At P the effort and the resistance are again equal. From P to B the resistance is greater than the effort. In other words, from A to N the work done by the steam is less than the resistance. This shows that the work represented by the area $A\,E\,M$ must have been done by the moving parts of the engine. From N to P the work done by the steam is greater than the resistance, and the excess of energy is absorbed by or stored in the moving parts. From P to the end of the stroke the work represented by the area $O\,F\,B$ is done on the crank pin by the moving parts.

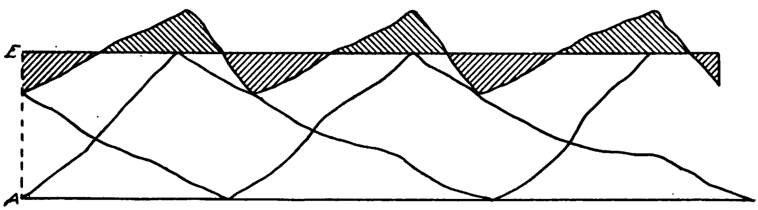


Fig. 55. Simultaneous Crank Effort Curves of Two Engines Acting at Right Angles to Each Other

It is known that energy is proportional to the square of the velocity from the formula

$$E = \frac{WV^2}{2g}$$

in which E is energy in foot pounds, W is weight in pounds, V is velocity in feet per second, and g is acceleration of gravity in feet per second^2. Hence as W and g remain the same, the velocity must be reduced when the moving parts are giving out energy and increased when receiving energy. Thus it is seen that the action of the crank pin is to move slowly, then more rapidly. The weight of the revolving parts of an engine is not sufficient to absorb sufficient surplus energy, hence a heavy flywheel is used.

In case there are two engines at right angles, two crank effort curves may be drawn, as shown in Fig. 55. The mean ordinate $A\,E$ is equal to the mean or constant resistance. There are two minimum and two maximum velocities in one stroke. The diagram shows

that the variation is much less than for a single cylinder, hence a lighter wheel may be used.

Calculations of Mass. The weight of the flywheel depends upon the character of the work done. For pumping engines and ordinary machine work the effort need not be as constant as for electric lighting. In determining the proper weight of a flywheel the diameter of the wheel must be known. If the wheel is too large, the high linear velocity of the rim will cause too great a centrifugal force and the wheel will not be safe. In practice, about 6,000 feet per minute is taken as the maximum linear velocity of cast-iron wheels. When made of wood and carefully put together the velocity may be taken as 7,000 to 7,500 feet per minute.

The linear velocity of a wheel is expressed in feet per minute by the formula $V=2\pi\ R\ N$, or $\pi\ D\ N$, in which V is velocity in feet per second, R is radius of wheel in feet, D is diameter of wheel in feet, and N is revolutions per minute.

Then if a wheel runs at 100 revolutions per minute, the allowable diameter would be obtained from the equation

$$6000=3.1416\times D\times 100$$

Therefore

$$D=\frac{6000}{3.1416\times 100}$$
$$=19.1 \text{ feet}$$

If a wheel is 12 feet in diameter the allowable speed is found to be

$$N=\frac{V}{\pi D}$$
$$=\frac{6000}{3.1416\times 12}$$
$$=159 \text{ revolutions per minute}$$

It is usual to make the diameter less than the calculated diameter.

Having determined the diameter, the weight may be calculated by several methods. There are many formulas to obtain this result given by various authorities, one formula being

$$W=\frac{C\times d^2\times b}{D^2\times N^2}$$

in which W is weight of rim in pounds; d is diameter of cylinder in inches; b is length of stroke in inches; D is diameter of flywheel in

feet; N is number of revolutions per minute; and C is a constant having a value which varies for different types of engines and for different conditions as follows:

Slide valve engines, ordinary work.	$C=$ 350,000
Corliss engines, ordinary work.	$C=$ 700,000
Slide-valve engines, electric lighting.	$C=$ 700,000
Automatic high speed engines.	$C=$1,000,000
Corliss engines, electric lighting.	$C=$1,000,000

EXAMPLE 1. Find the weight of a flywheel rim for an automatic high speed engine used for electric lighting. The cylinder is 24 inches in diameter; the stroke is 2 feet. It runs at 300 revolutions per minute, and the flywheel is to be 6 feet in diameter.

SOLUTION.

$$W=\frac{1000000\times(24)^2\times24}{36\times90000}$$
$$=4266 \text{ pounds}$$

EXAMPLE 2. A plain slide valve engine for electric lighting is 20 inches $\times$ 24 inches. It runs at 150 revolutions per minute. The flywheel is to be 8 feet in diameter. What is the weight of its rim?

SOLUTION.

$$W=\frac{700000\times400\times24}{64\times22500}$$
$$=4666 \text{ pounds}$$

The weight of a flywheel is considered as being in the rim. The weight of the hub and arms is simply extra weight. Then, if the weight of the rim and its diameter be known, the width of the face and thickness of the rim can be found. Assume the given diameter to be the mean of the diameter of the inside and outside of the rim. Let b equal width of face in inches; t equal thickness of rim in inches; d equal diameter of flywheel in inches; and .2607 equal weight of 1 cubic inch of cast iron. Then

$$W=.2607\times b\times t\times\pi d$$
$$=b\times t\times.819\,d$$

EXAMPLE 3. Suppose the rim of a flywheel weighs 6,000 pounds, is 9 feet in diameter, and the width of the face is 24 inches. What is the thickness of the rim?

SOLUTION.

$$t=\frac{W}{.819\,db}$$
$$=\frac{6000}{.819\times108\times24}$$
$$=2.83 \text{ inches}$$

In this case the rim would probably be made $2\frac{13}{16}$ inches thick. The total weight, including hub and arms, would probably be about 8,000 pounds.

GOVERNOR

The load on an engine is never constant, although there are cases where it is nearly uniform. While the engine is running at constant speed, the resistance at the flywheel rim is equal to the work done by the steam, disregarding friction. If the load on the engine is wholly or partially removed and the supply of steam continues undiminished, the force exerted by the steam will be in excess of the resistance. Work is equal to force multiplied by distance; hence, with constant effort, if the resistance is diminished, the distance must be increased. In other words, the speed of the engine will be increased, and the engine will "race." Also, if the load increases and the steam supply remains constant, the engine will "slow down."

It is evident, then, that if the speed is to be kept constant some means must be provided so that the steam supply shall at all times be exactly proportional to the load. This is accomplished by means of a governor.

Methods of Action. Steam-engine governors act in one of two ways (1) they may regulate the pressure of steam admitted to the steam chest, or (2) they may adjust the speed by altering the amount of steam admitted. Those which act in the first way are called *throttling governors,* because they throttle the steam in the main steam pipe. Those of the latter class are called *automatic cut-off governors,* since they automatically regulate the point of cut-off.

Theoretically, the method of governing by throttling the steam causes a loss in efficiency, but the throttling superheats the steam, thus reducing cylinder condensation. By the second method the loss in efficiency is very slight, unless the ratio of expansion is already great, in which case shortening the cut-off causes an increasing cylinder condensation.

Control by Centrifugal Force. In most governors of the throttling type and those applied to Corliss engines, centrifugal force counteracted by some other force is employed. A pair of heavy masses (usually iron balls or weights) are made to revolve about a spindle, which is driven by the engine. When the speed increases, the centrifugal force increases and the balls tend to fly outward, that is, they revolve in a larger circle. The controlling force, which is usually gravity or springs, is no longer able to keep the balls in

their former path. When, therefore, the increase is sufficiently great, the balls in moving outward act on the regulator, which may throttle the steam or cause cut-off to occur earlier.

With the throttling governor, a balanced throttle valve is placed in the main steam pipe leading to the valve chest. If the engine runs faster than the desired speed, the balls are forced to revolve at a higher speed. The increase in centrifugal force will cause them to revolve in a larger circle and in a higher plane. By means of levers and gears, the spindle may be forced downward, thus partially closing the valve. The engine, therefore, takes the steam at a low pressure, and consequently the speed falls slightly.

Similarly, if the load is increased, the engine slows down, causing the balls to drop and open the valve more widely; steam at higher pressure is then admitted and the speed is increased to the regular number of revolutions.

With the Corliss or other four-valve engines, the governor acts differently. Instead of throttling the steam in the steam pipe, the governor is connected to the releasing gear by rods. An increase of speed causes the releasing gear to unhook the disengaging link earlier in the stroke. This causes earlier cut-off, which of course decreases the power and speed, since the amount of steam admitted is less. If for any reason the load increases, the governor causes the valves to be held open longer. The cut-off, therefore, occurs later in the stroke.

Pendulum Governor. One of the most common forms of governor is similar to that invented by James Watt. It is called from its appearance the pendulum governor and is illustrated in principle in Fig. 56. To consider the theory of the pendulum governor, the masses of the balls are assumed to be concentrated at their centers and the rods are made of some material having no weight.

When the governor is revolving about its axis at a constant speed, the balls revolve in a circle having a radius r. The distance from this plane to the intersection of the rods, or the rods produced, is called the height and is equal to h.

If the balls revolve faster, the centrifugal force increases, r becomes greater, and h diminishes. The mathematical expression for centrifugal force is

$$F = \frac{Wv^2}{gr}$$

in which F is force in pounds; W is weight of one ball in pounds; v is velocity in feet per second; g is acceleration due to gravity; and r is radius in feet. From the above equation it is seen that force varies inversely as the radius.

While the pendulum is revolving, centrifugal force acts horizontally outward and tends to make the balls fly from the center; and the action of gravity tends to make the balls drop downward. In order that the balls shall revolve at a certain height, the moments of these two forces about the point of suspension must be equal, or

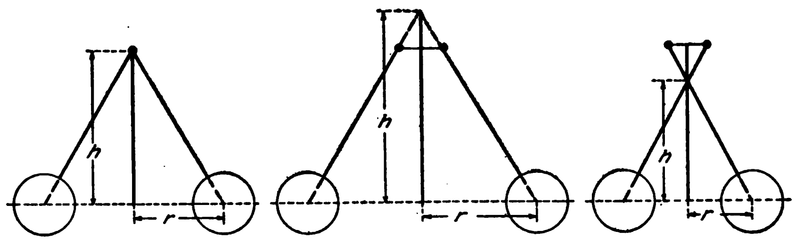


Fig. 56. Diagrams Showing Action of Pendulum Governor

the weight of the balls multiplied by their distance from the center must equal the centrifugal force multiplied by the height, or

$$W \times r = F \times h$$

from which

$$\frac{h}{r} = \frac{W}{F}$$

Substituting value of F just given, we have

$$\frac{h}{r} = \frac{W}{\frac{Wv^2}{gr}}$$

$$= \frac{gr}{v^2}$$

Therefore,

$$h = \frac{gr^2}{v^2}$$

Now since v, the linear velocity of a point revolving in the circumference of a circle, is expressed as $2\,\pi\,r\,N'$ feet per second, where N'

is revolutions per second, this value may be substituted in the above formula, giving

$$h=\frac{g\,r_2}{4\,\pi^2 r^2 (N')^2}$$

$$=\frac{g}{4\,\pi^2\,(N')^2}$$

and since the values of g and π are known, the formula may be written

$$h=\frac{32.16}{4\times 3.1416^2\times(N')^2}$$

$$=\frac{.8146}{(N')^2}\text{ feet}$$

$$=\frac{9.775}{(N')^2}\text{ inches}$$

If it is desired to use N, the r.p.m., instead of N', the r.p.s., the former may be substituted in the formula by multiplying the fraction by $\overline{60}^2$, or 3600, giving

$$h=\frac{2932.56}{N^2}\text{ feet}$$

$$=\frac{35190.7}{N^2}\text{ inches}$$

From the above formula it is evident that the height is independent of the weight of the balls or the length of the rod, depending entirely upon the number of revolutions. The height varies inversely as the square of the number of revolutions.

The ordinary pendulum governor is not isochronous, that is, it does not revolve at a uniform speed in all positions, the speed changing as the angle between the arms and spindle changes.

Fly-Ball Governor. The early form consisted of two heavy balls suspended by links from a pin connection in a vertical spindle, as shown in Figs. 57 and 58. The spindle is caused to revolve by belting or gearing from the main shaft, so that as the speed increases, centrifugal force causes the balls to revolve in a circle of larger and

larger diameter. The change of position of these balls can be made to affect the controlling valves so that the admission or throttling will vary with their position. With this governor it is evident that for a given speed of the engine there is but one possible position for the governor, consequently one definite amount of throttling or one point of cut-off, as the case may be. If the load varies, the speed of the engine will change. This causes the position of the governor balls to be changed slightly, thus altering the pressure. But in order that the pressure or cut-off shall remain changed, the governor balls must stay in their new position. That is to say, the speed of the engine

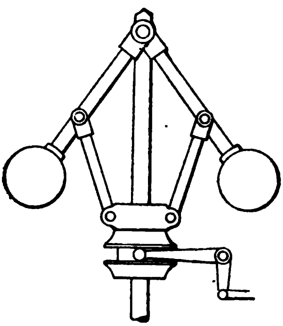


Fig. 57. Simple Type of Fly-Ball Governor

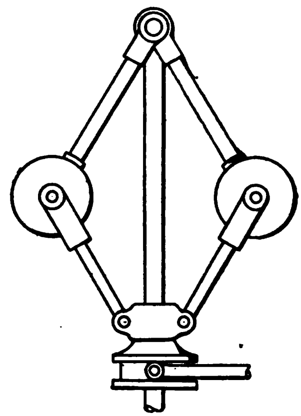


Fig. 58. Later Type of Fly-Ball Governor

must be slightly changed. Thus with the old ball governors there was a slightly different speed for each load. This condition has been greatly improved by various modifications until now such governors give excellent regulation.

While the engine is running with a light load, the valve controlled by the governor will be open just enough to admit steam at a pressure that will keep the engine running at a given speed. Now if the engine is heavily loaded, the throttle valve must be wide open. The change of opening is obtained by a variation in the height of the governor, which is caused by a change of speed. Thus it is seen that the governor can control the speed only within certain limits which are not far apart. The difference in the extreme heights of the governor must be sufficient to open the throttle its entire range. In

TABLE II

Heights of Governor for Different Speeds of Engine

Number of Revolutions Per Minute	Height in Inches	Variation of Height in Inches 4 Per Cent
250	.563	.0225
200	.879	.035
175	1.149	.046
150	1.564	.062
125	2.252	.090
100	3.519	.140
75	6.256	.250
50	14.076	.563

most well-designed engines, equipped with a throttling governor, the speed will not vary more than 4 per cent, that is, 2 per cent above or below the mean speed.

From the formula $h=\frac{35190.7}{N^2}$, the heights corresponding to given speeds can be computed as shown in the second column of Table II. The third column is the variation in height for a speed variation of 4 per cent or 2 per cent either above or below the mean.

Disadvantage of Ordinary Fly-Ball Type. From Table II it will be seen that for a considerable variation of speed there is but slight variation in the height of the governor, this being too small to control the cut-off or throttling mechanism throughout the entire range. Also for high speeds the height of the governor is so small that it would be difficult to construct it.

Other disadvantages of the fly-ball governor are as follows: It is apparent that the valves must be controlled by the weight of the governor balls. In large engines this requires very heavy balls in order to quickly overcome the resistance of the valves. But these large balls have considerable inertia and will therefore be reluctant to change their speed with that of the engine. The increased weight will also increase the friction in the governor joints and the cramping action existing when the balls are driven by the spindle will increase this friction much further. All these things tend to delay the action of the governor, so that in all large engines the old-fash-

ioned governor became sluggish. The balls had to turn slowly because they were so heavy; this was especially troublesome in high-speed engines.

Porter Improved Type. To remedy these defects the weighted or Porter governor, Fig. 59, was designed. It has a greater height for a given speed, and the variation in height for a given variation of speed is greater and, consequently, more sensitive. By increasing this variation in height, the sensitiveness is increased. Thus, if a governor running at 50 revolutions has a variation in height of .57 inch, it is not as sensitive as one having a variation of 1 inch for the same speed.

In the weighted governor, the weight is formed so that the center of gravity is in the axis. It is placed on the spindle and is free to revolve. The weight adds to the weight of the balls, and thus increases the moment of the weight. It does not, however, add to the centrifugal force, and hence the moment of this force is unchanged. It may then be said that the weight adds effect to the weight of the governor balls but not to the centrifugal force, and as a consequence the height of the governor for a given speed is increased. If W equals the weight of the ball as before, and W' equals one-half the added weight, the equated moments are

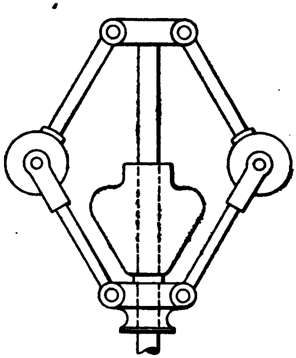


Fig. 59. Porter Improved Type of Fly-Ball Governor

$$(W+W')\,r = Fh$$

Substituting for F its value obtained from the formula, p. 107, we have

$$(W+W')\,r = \left(\frac{Wv^2}{gr}\right) h$$

$$h = (W+W')r\left(\frac{gr}{Wv^2}\right)$$

$$= \frac{(W+W')r^2g}{W\times 4\pi^2 r^2 (N')^2}$$

$$= \frac{(W+W')}{W}\times\left(\frac{g}{(4\pi^2 (N')^2}\right)$$

Since it is known that

$$\frac{g}{4\pi^2(N')^2}=\frac{.8146}{(N')^2}$$

$$h=\left(\frac{W+W'}{W}\right)\times\frac{.8146}{(N')^2}$$

Hence the height of a weighted governor is equal to the height of a simple pendulum governor multiplied by $\left(\frac{W+W'}{W}\right)$, or $\left(1+\frac{W'}{W}\right)$.

For instance, if the height of a simple pendulum is 10 inches and

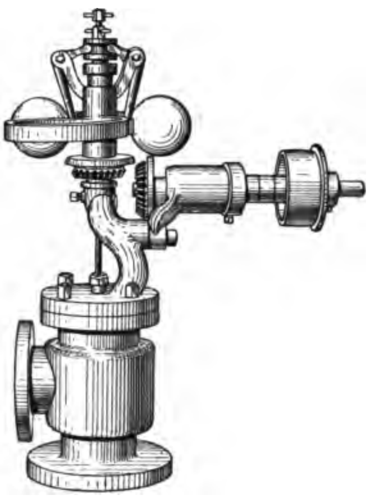


Fig 60. Waters Spring Type of Fly-Ball Governor

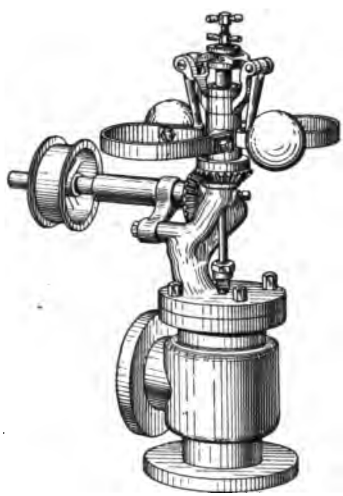


Fig. 61. Waters Governor with Safety Stop

the weight of the balls equal to the added weight, the height of the weighted governor will be

$$h=\left(1+\frac{1}{1}\right)\times 10$$

$$=2\times 10$$
$$=20$$

Thus it is evident that if a weight equal to the combined weight of the balls is added, the height of the governor will be doubled. If the belt driving the governor slips off or breaks, the balls will

drop, with the result that the engine will "run away." To diminish this danger many governors are provided with some kind of safety stop which closes the valve when the governor loses its normal action. Usually a trip is provided which the governor does not touch in its normal positions, but which will be released if the balls drop down below a certain point.

Spring Type. In many cases a spring is used in place of the weight. This type of governor is frequently used on throttling engines, and it consists of a pendulum governor with springs added to counteract the centrifugal force of the balls. Thus the height and sensitiveness are increased. Fig. 60 shows the exterior view of a Waters governor and Fig. 61 shows the same governor having the safety stop. In this governor the weights are always in the same plane, the variation in height being due to the action of the bell-crank levers connecting the balls and spindle. When the balls move outward, the spindle moves downward and tends to close the valve. The governor balls are caused to revolve by means of a belt and bevel gears. The valve and seat are shown in section in Fig. 62. The valve is a hollow cylinder with three ports through which steam enters. The seat is made in four parts, that is, there are four edges that the steam passes as it enters the valve. The valve, being cylindrical and having steam on both sides, is balanced, and because of the many openings only a small travel is necessary.

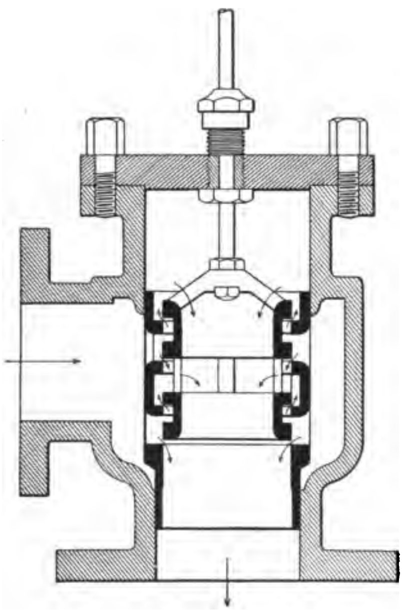


Fig. 62. Section of Valve and Valve Seat of Waters Governor

Shaft Governor. Usually some form of pendulum governor is used for throttling engines. For governing an engine by varying the point of cut-off, shaft governors are generally used, although the Corliss and some other engines use pendulum governors for this pur-

pose. Cut-off governors, which are called shaft governors because they are placed on the main shaft, are made in many forms, but their essential features are the same. Two pivoted masses or weights are arranged symmetrically on opposite sides of the shaft and their tendency to fly outward when the speed increases is resisted by springs. When in action the outward motion of the weights causes the admission valve to close earlier, and the inward motion causes it to close later. This change is effected by altering the position of

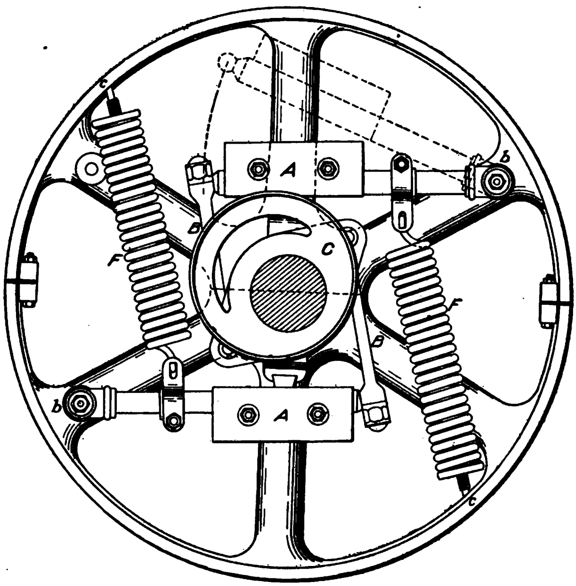


Fig. 63. Diagram Showing Action of Buckeye Shaft Governor

the eccentric, either by changing the eccentricity or the angular advance.

Shaft governors are made in a great variety of ways, no two being exactly alike. If the principles of a few types are understood, it is easy to understand others.

Buckeye Type. The valve of the Buckeye engine is hollow and of the slide valve type. The cut-off valve is inside. The change of cut-off is due to the alteration of the angular advance, the arrangement of the parts which effect this alteration being shown in Fig. 63. A wheel which contains and supports the various parts of the gov-

ernor is keyed to the shaft. Two arms, having weights *A A* at the ends, are pivoted to the arms of the wheel *b b*. The ends having the weights are connected to the collar on the loose eccentric *C* by means of rods *B B*.

When the weights move to the position indicated by the dotted lines, the eccentric is turned on the shaft about a quarter of a revolution in the direction in which the engine runs, that is, the eccentric is advanced, or the angular advance is increased; this makes cut-off occur earlier, as shown by the table presented in "Valve Gears." If the engine had a single plain slide valve, the variation of the angular

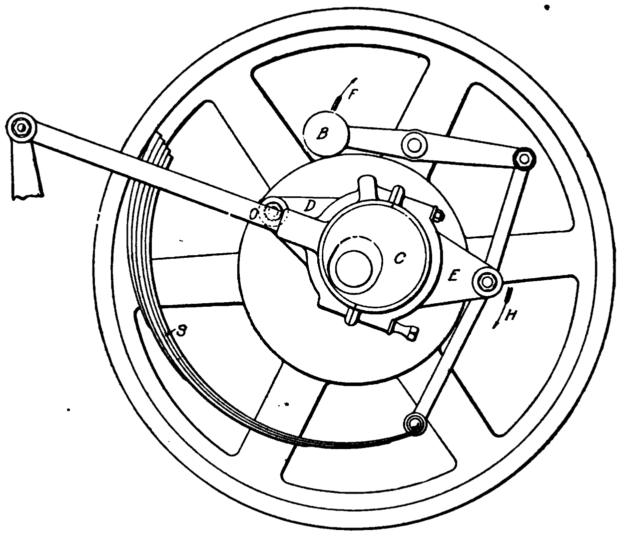


Fig. 64. Diagram Showing Action of Straight-Line Type of Shaft Governor

advance would produce too great a variation of lead; but as this engine has a separate valve for cut-off, admission is not altered by the cut-off valve.

The springs *F F* balance the centrifugal force of the weights; the weights *AA* are varied to suit the speed; and the tension on the springs is altered by means of the screws *c c*. Auxiliary springs are added in order to obtain the exactness of regulation necessary for electric lighting. These springs tend to throw the arms outward, but act only during the inner half of this movement.

Straight-Line Type. Fig. 64 shows the governor of the Straight-line engine. It has but one ball *B*, which is linked to the spring *S*

and to the plate *DE*, on which is the eccentric *C*. When the ball flies outward in the direction indicated by the arrow *F*, the eccentric is shifted about the pivot *O*, the links moving in the direction of the arrow *H*. The ball is heavy and at a considerable distance from the center, hence it has a great centrifugal force and the spring must be stiff. The governor of the Buckeye engine alters the cut-off by changing the angular advance, while the Straight-line engine governor changes the travel of the valve. The latter type of valve is very common.

Inertia Form. The well-known Rites inertia governor, Fig. 65, is a form of shaft governor largely used for certain types of engines. This governor regulates the speed of the engine by shifting the eccentric, thus changing the valve travel and increasing or decreasing the angular advance, depending on the speed conditions. It differs in its operation from the centrifugal shaft governor previously considered, in that it makes use of the inertia of two large weights instead of centrifugal force. To understand this action, it first becomes necessary to know something about its construction.

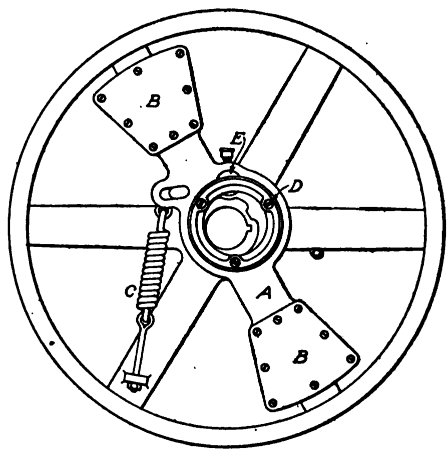


Fig. 65. Diagram of Rites Inertia Type of Shaft Governor

The governor consists essentially of a heavy arm *A* pivoted at *E* to the flywheel. This arm carries two heavy weights at *B*. The eccentric *D* is fastened to the arm by three countersunk screws, as shown, and moves with reference to the engine shaft whenever the weights *B* cause the arm *A* to move about its pivot point. Fastened to the flywheel arm and the governor arm *A* is the spring *C*, which brings the arm *A* back to its normal position when the engine is not operating. This spring also has certain other functions to perform in the operation of the governor.

The action of the governor is such that the valve experiences very much the same movement as in the centrifugal governor. As

the engine speeds up, the tendency of the heavy arm *A* is to lag behind the flywheel. This lagging action controls the position of the eccentric so that the valve travel is reduced, thus limiting the amount of steam that enters the cylinders. If, after the engine is operating at a uniform rate of speed, an increase of load suddenly occurs, the motion of the engine shaft and flywheel will be slightly retarded and the engine will commence to "slow down." On account of the energy stored up in the governor arm and the weights *BB*, they will not be so quickly affected, hence the governor will be moving slightly faster than the shaft. As a result the eccentric position with reference to the shaft will be changed, and the valve travel increased, thus permitting more steam to enter the cylinder, increasing the power commensurate with the added load. If for any reason the engine takes a sudden spurt in speed, the tendency of the governor is to fall backward, so to speak; and if the engine is suddenly slowed down for any cause, the tendency of the governor is to plunge forward; hence the valve travel is shortened or lengthened according to which action takes place. This type of governor gives very close regulation when properly constructed.

ERECTION AND OPERATION OF STEAM ENGINES

The limited scope of this work will not permit of an exhaustive study of these two important details—the erection and operation of steam engines; only the general principles governing each will be pointed out.

ERECTION

Foundations. When about to erect an engine the first requisite is the foundation, the character of which will, of course, depend upon the type and the size of the engine. It should be built according to plans submitted by the engine builders, no changes of material consequence being made without the approval of the builders. It should be neither connected with nor in close proximity to any supporting column or columns of the building, as vibrations of the engine will be transmitted to the building which might prove to be disastrous. The foundation should be built upon a solid bottom, but if this is not obtainable at the depth required by the foundation plans, the base of the foundation should be extended in all directions in

order that the bearing surface may be increased. In the case of the horizontal engines the nearer the center of gravity of the foundation is placed to the center line of the engine, the more effective will be the foundation. In such cases, therefore, it is preferable to have an extended bearing surface rather than one of considerable depth. The foundation bolts and washers should be carefully located in accordance with the furnished plans. A space of one inch or more should be left around each of the foundation bolts. This may be obtained by using pieces of short iron pipe or old boiler tubes around the bolts, care being taken that they do not extend above the foundation, so as to prevent the proper tightening of the bolts after the engine is placed in position. After the engine is properly set, the space left around the foundation bolts should be filled with the best cement mortar, so as to insure their permanency. The foundation should be a solid one and built of brick, stone, or concrete.

Brick. When brick is used, a hollow square effect may be constructed and the open space filled with a mixture of concrete, consisting of one part cement and three parts sand and gravel.

Concrete. When making a concrete foundation, suitable forms must first be constructed to receive the concrete. . Crushed stone or clean gravel or both may be used, care being taken to wash the gravel free of all clay. A good mixture for ordinary foundations is one having the proportions: $1:2\frac{1}{2}:5$. That is, 1 barrel, or 4 bags, cement, $2\frac{1}{2}$ barrels, or 9.5 cubic feet, of sand, and 5 barrels, or 19 cubic feet, of gravel or stone. If the foundation is to be waterproof, careful consideration must be given to the proportioning of the mixture. If the foundation covers considerable area and is not very deep, the mixture should be richer in cement; if, however, the foundation is very deep, a poorer mixture may be used at the bottom and a richer one near the top.

The cement, gravel or stone, and sand should first be thoroughly mixed in the dry state and the water added while the mixing process continues until the mass is well mixed and thoroughly wet. After the mixing is complete, the concrete should be laid in layers from 6 to 9 inches deep and well rammed until solid. The ramming of the concrete is an absolute necessity in order that a solid foundation may be secured.

When the foundation has been completed in accordance with the furnished plans, sufficient time must elapse before any machinery

is placed thereon in order to insure a proper setting of the cement. When the concrete has set sufficiently, it should be inspected to see that no omissions or errors have been made, after which the engine may be unpacked and prepared for setting. If the foundation is a large one, an inspector should be on hand at all times to follow the work and see that no errors are made.

Setting the Engine. Upon the accuracy and thoroughness of the setting of the engine, in a large measure depends its successful operation as to smoothness and efficiency of running. In this process there are a great many things to be considered. First, the base and sub-base must be carefully cleaned and set in position. Next, the crank shaft, cylinders, piston, crosshead valves, and other details must be carefully placed in position and alignment made according to the plans of the builders. As all of these details require skill, an inexperienced person should not attempt the setting up of an engine. It is always preferable, when possible, to obtain an experienced man from the engine builders.

Installation of Attachments. In addition to the erection and setting of the engine proper there are various attachments and auxiliaries that require care and skill in their proper installation. The steam and exhaust piping as well as the cylinder drainage should be carefully attended to. The piping should be of ample size, all bends should be easy, and gate valves should be used whenever possible. The piping should have a gradual fall from the boiler to the engine, at or near which should be placed a separator.

Separator. The separator should be of approved design, and care must be taken to carefully provide for drainage in order to insure the removal of the water, otherwise the separator might form a reservoir for water and thus endanger the engine more with its use than without. In addition to being a safeguard against water hammer, when properly attached, the separator also improves the steam economy of the engine, since it removes the most of the entrained moisture which is carried from the boiler through the steam pipes.

Exhaust Pipes. The exhaust pipes should be of ample area to take care of all exhaust steam, and safeguards should be used to insure no backing up of the condensed exhaust into the cylinders. To this end, sharp bends should be avoided and gate valves should be used

if valves are necessary, as by their use the area of the pipe is less reduced than by other forms of valves. Check valves should be avoided whenever possible.

Cylinder Drains. The cylinder drains should be of sufficient area to care for all condensed steam in the cylinders and so attached to the cylinders and the exhaust pipe or receiver that no pockets will be formed for the accumulation of water. In the case of compound engines the cylinder drains of the high and the low pressure cylinders should not be connected together, but separately connected to the exhaust or other main drain. In condensing engines the cylinder drains should always go into the exhaust drain if it is low enough to admit of proper drainage.

OPERATION

Let us now turn our attention to the operation and management of an engine. It should be borne in mind that many suggestions as to the proper alignment and adjustment of bearings, the adjustment of valves, and the consideration given lubrication will be applicable both to the first setting up of the engine and also to the daily operation afterwards.

Competent Engineer a Requisite. The operation of an engine should be committed to a careful, skillful, and reliable man. This is especially true in the case of modern well-equipped plants which represent quite an outlay of capital. In many of the smaller plants, however, not much attention is given to the matter and we find, as a result, men holding positions as operators who know very little about their business. Under such conditions the plants are seldom operated efficiently.

As a suggestion of some of the duties of a man in charge of a modern plant, which also suggest the amount of judgment and experience required, the following general instructions are presented.

Care of Bearing Caps. The caps on the main bearings should always have sufficient liners underneath to enable the nuts on the bearing studs to draw the cap down tightly upon them and not pinch the shaft, which should be free to revolve in its bearings without unnecessary play.

The caps should be removed occasionally as conditions demand in order to clean out the oil grooves which are chipped in the babbitt

metal, as the passages may become clogged with dirt or other foreign matter.

Adjustment of Connecting Rod Box. In adjusting the connecting rod box at the crank pin end, the same general rules should be observed regarding the liners under the cap—the large nuts drawn solidly upon it, the small nuts firmly jammed and the cotter pins placed in position. The adjustment of the box should then be tested with a lever about 12 inches in length, the adjustment being so made that with a lever of this length the operator can easily move the end of the connecting rod sufficiently to take up the side play between the flanges on the crank pin and the end of the box. The adjustment should never be made so close that this side movement can not be observed.

The adjustment of the connecting rod box at the crosshead pin should be made by placing the crank on the center nearest the cylinder; then with a wrench provided for that purpose, slack off both wedge screws at the upper and lower sides of the connecting rod, and draw the wedge up until it is solid against the box; then slack off one screw about a sixth of a turn, and draw up the other so as to firmly lock the wedge.

Lining Up Crosshead. The crosshead should be lined up between the guides, while disconnected from the connecting rod. When in this condition the crosshead should be so lined that it can be easily pulled from one end of the guides to the other with a short lever.

The crosshead should never be run very close, and should always be free enough to allow long and continuous runs without heating the guides to the degree that they would be uncomfortably warm to the touch.

When making any adjustments of the crosshead, the operator should assure himself that the lock nut which prevents the piston rod turning in the boss of the crosshead is securely placed.

Adjusting Eccentric Strap. The eccentric strap adjustment is made by liners placed between the halves of the strap and double nutted bolts. When adjustment is necessary, the other end of the eccentric rod should be disconnected and, after drawing up the strap bolts, it should be tested by giving the strap a half revolution about the eccentric. If it is found that the friction between the strap and the eccentric is sufficient to support the weight of the rod, the bolts

should be loosened and liners replaced until the strap moves freely without lost motion. The double nuts should then be locked and the cotter pins replaced in the ends of the bolts.

Governor. The governor should be adjusted to meet the different conditions of speed and steam pressure and the degree of regulation required. As governors differ so much in design and detail of construction, it is not possible to give any general rule for their adjustment. The operator, if desired, can usually obtain instructions from the engine builder for the particular type of governor in question.

Valve Setting. As a discussion of the setting of the valves and their adjustment for wear will be found given in "Valve Gears," no consideration of the subject will be presented here.

Lubrication. The lubrication of a steam engine, and especially of high speed engines, is a very important consideration with both the designer and the operator, for it is upon proper lubrication that they must largely depend for a constant and satisfactory operation. The designer must, therefore, provide ample and efficient facilities for lubricating the bearings, cylinders, and valves, whereas the operator must use discretion in selecting his lubricants and the amount to use after selection has been made.

Choice of Oils. It might be said that only the best oils should be used. Cheap oils are usually considered expensive at any cost and should be avoided as they promote excessive wearing of the parts—causing noisy operation--and may cause serious cutting of the cylinders. There are two general classes of liquid lubricants now in the market, namely, mineral and animal oils. There is also a compounded lubricant which is made up of about 5 to 15 per cent of animal matter and the balance of mineral oil. This compound makes a very efficient lubricant for some classes of service, as it withstands the action of the condensation and adheres to the surface of the cylinders, thus giving better results than larger quantities of mineral oil.

In plants where open heaters are used and where the exhaust steam is condensed and used for boiler feed water, the compounded oils can not be used, on account of the danger of the animal matter getting into the boilers and causing considerable trouble. In such cases mineral oil must be used, although it may require considerable more mineral than compounded oil to accomplish the lubrication.

Solid Lubricants. Several solid lubricants are used, such as graphite, metalline, soapstone, and fiber graphite.

Graphite when mixed with certain oils is well adapted for heavy pressures. It is especially good for heavy pressures and low velocities. Under conditions which require a large amount of cylinder oil, a small amount of crystal or flake graphite may be used with good results. Care must be exercised, however, if the exhaust steam is used for feed water, as the graphite may get into the boilers and cause inconvenience and perhaps serious trouble.

Metalline is a solid compound containing graphite. It is made in the form of solid cylinders, which are fitted to the holes drilled into the surface of the bearing. When a bearing is thus fitted, no other lubricant is necessary.

Soapstone in the form of powder and mixed with oil or fat is sometimes used as a lubricant. Soap mixed with graphite or soap-stone is often used where wood is in contact with wood or iron.

A preparation called *fiber graphite* is used for self-lubricating bearings. It is made of finely divided graphite mixed with fibers of wood. It is pressed in molds and afterwards fitted to bearings.

For great pressure at slow speed, graphite, lard, tallow, and other solid lubricants are suitable. If the pressure is great and the speed high, castor, sperm, and heavy mineral oils are used.

For low pressure and high speed, olive, sperm, rape, and refined petroleum give very satisfactory results.

In ordinary machinery, heavy mineral and vegetable oils and lard oil are good. The relative value of various lubricants depends upon the prevailing conditions. Oil that is suitable for one place might not flow freely enough for another.

The quality of oil is of great importance. In many branches of industry it is imperative that the machinery run as perfectly as possible. On this account and because of the high cost of machinery, only first class oil should be used. The cylinder oil especially should be high grade, because the valves, piston, and piston rods are the most delicate parts of the engine.

Qualities of a Good Lubricant. From the foregoing brief discussion of lubricants it will be evident that they must possess certain qualities which may be enumerated as follows:

The lubricant must be sufficiently fluid, so that it will not in itself make the bearing run hard.

It must not be too fluid or it will be squeezed out from between the bearing surfaces. If this happens, the bearing will immediately heat and begin to cut. The heating will tighten the bearing and increase the pressure and the cutting.

It must not gum or dry when exposed to the air.

It must not be easily decomposed by the heat generated. If it should be decomposed, it might form substances which would be injurious to the bearings.

It must not take fire easily.

It must contain no acid and should form no acid in decomposing, as acids corrode the bearings.

Both mineral and animal oils are used as lubricants. Formerly animal oils were used entirely, but they were likely to decompose at high temperatures and form acids. It is important in using high pressure steam to have "high test oils," that is, oils which will not decompose or volatilize at the temperature of the steam. It was the difficulty of getting such oils which made great trouble when superheated steam was first used. Mineral oils will stand high temperatures very readily, and even if they do decompose, they form no acids.

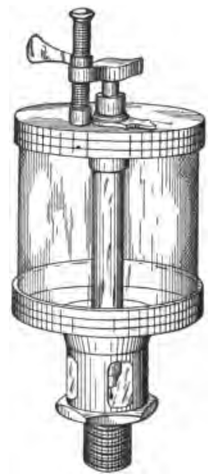


Fig. 66. Standard Type of Simple Oil Cup

Common Oilers. Engines are lubricated by means of oil cups and wipers placed on the bearings wherever required. They are made in many forms. Formerly, the oil cup was made with a tube extending through the oil. A piece of lamp wick or worsted leads from the oil in the cup to the tube. Capillary attraction causes the oil to flow continuously and drip down the tube. When not in use, the lamp wick should be withdrawn. This type of oil cup is now seldom used.

The oil cup shown in Fig. 66 is simple and economical. The opening of the valve is regulated by an adjustable stop. The oil may be seen as it flows drop by drop. The cylindrical portion is made of glass, so that the operator can see how much oil there is in the cup without opening it.

A form of wiper crank pin oiler is shown in Fig. 67. The oil cup is attached to a bracket. The oil drops from the cup into the sheet of wicking or wire cloth and is removed at each revolution of the crank pin by means of the cup which is attached to the end of the connecting rod. This form of oiler works very satisfactorily at slow speeds.

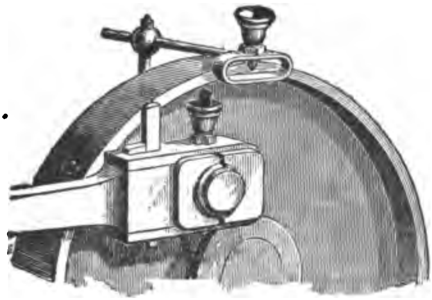


Fig. 67. Form of Wiper Crank Pin Oiler

Centrifugal Oilers. Fig. 68 shows a centrifugal oiling device which operates very satisfactorily at all speeds. The oil flows from the oil cup through the tube to the small hole in the crank pin by centrifugal force. It reaches the bearing surface by means of another small hole.

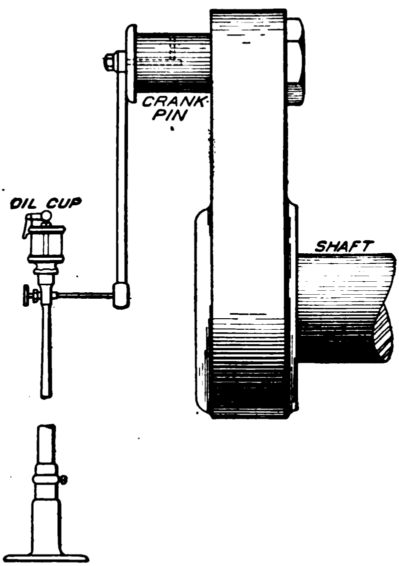


Fig. 68. Centrifugal Oiler

Cylinder Lubrication. In oiling the valve chest and the cylinder, the lubricant must be introduced against the pressure of the steam. This may be done in several ways, in each of which it is introduced into the steam before it reaches the valve chest and is carried by the steam to the surfaces to be lubricated.

By Oil Pumps. The oil may be forced into the steam pipe by a small hand pump or, in large engines, by an attachment from the engine itself. The supply of oil is, of course, intermittent if the pump is driven by hand, but continuous and economical if driven by the engine.

By Sight-Feed Lubricators. The most common device for feeding oil to the cylinder is that which introduces the oil drop by drop into the steam when it is in the steam pipe or steam chest. The oil

becomes vaporized and lubricates all the internal surfaces of the engine.

Fig. 69 shows the section of a sight-feed lubricator, which must be placed on the steam supply pipe in a vertical position above the throttle. The reservoir *O* is filled with oil. The pipe *B*, which connects with the steam pipe, is partly filled with condensed steam which flows down the small curved pipe *E* to the bottom of the chamber *O*. A small portion of the oil is thus displaced and flows from the top of the reservoir *O* down the tube *F* by the regulating valve *D*, and up through the glass tube *S*, which is filled with water. It enters the main steam pipe through the connection *A*. The gauge glass *G* indicates the height of water in the chamber *O*. To fill the lubricator, close the regulating valve *D* and the valve in pipe *B*; the oil chamber can thus be drained through the cock *C*, and filled. If the glass *S* becomes clogged, it may be cleaned by closing valve *D* and opening the small valve *H*. This will allow the steam to blow through the glass. After cleaning close valve *H* and allow glass *S* to become filled with water before opening the feed valve. The amount of oil fed to the cylinder can be regulated by opening the valve *D* the proper amount. The exact quantity of oil necessary for the engine is not easily determined. For ordinary sizes it is from one to four drops per minute, depending on the conditions.

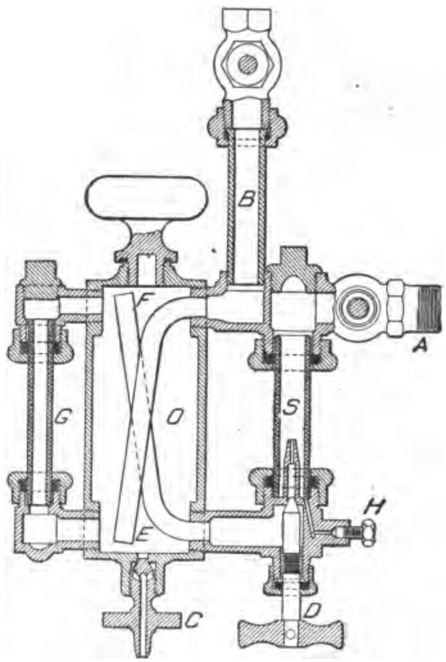


Fig. 69. Section of Sight-Feed Lubricator

Instructions for Proper Lubrication. In slow speed engines it is not a difficult matter to attend to the oiling; all the parts are moving slowly and can be readily examined and oiled. Many high speed

engines run so fast that it is impossible to examine the various parts, and special means must be provided for lubrication. It is especially important in high speed engines that there should be no heating.

In order to avoid the danger of neglecting to oil a bearing of a high speed engine, it is customary to have all the bearings oiled from one central source. All the oil is supplied to one reservoir, from which pipes lead to all bearings. If this is not done, large oil cups are used, as a rule, so that oiling need not be attended to as frequently.

In some high speed engines the moving parts are enclosed and the crank runs in a bath of oil. This secures certain oiling and is very effective. All the bearings may be inside this crank case, so that all are oiled in this way. It is thus impossible for a careless operator to overlook one point and so endanger the whole engine.

Starting the Engine. Before starting an engine, the oil cups should be started feeding, grease cups screwed down, and the governor and other parts of the valve gear oiled. The cylinder lubricator should be started before the engine so that the oil passages will contain oil. The cylinder drain cocks should be open so that any condensed steam in the cylinder will be removed without injury to the cylinder.. These precautions having been observed, the throttle may be opened slowly and the engine started and gradually brought to the required speed.

After starting the engine, notice should be taken of the governor and all the lubricating apparatus to see that each is properly performing its function.

When the engine is to operate condensing, the condenser should be started first, if it is in such a position that the water in the exhaust can drain into it. If the condenser is above the engine and no means are provided for removing the water, the engine should be started non-condensing. When a jet condenser is used, the quantity of injection water should be increased as the load is increased; the amount being determined by the conditions of the vacuum and temperature of the discharge water, which should be from 100° to 110°F. If the water is colder than this, it would denote that more injected water is being used than is required.

The foregoing suggestions and indicated precautions are only a few of the more important things that will arise in the course of the erection, setting, and operation of an engine. The one performing

these various duties must at all times exercise good judgment and act according to what his past experiences and that of others have taught under similar circumstances.

ENGINE SPECIFICATIONS

Selecting an Engine. The engineer who has the responsibility of selecting an engine for a given class of service has no small task to perform, if he carefully analyzes all the factors entering into the problem. If the installation contemplated is to be an extensive or expensive one, expert advice should be solicited. Since this is not always to be had, a few suggestions will be given as to how best to proceed when one has to specify an engine for a given service. Consider for the time being that an expert consulting engineer is not available and a rather inexperienced person, or non-technical man, who knows little about the theoretical questions that should be given consideration, has to select the engine. In this case the most satisfactory procedure to follow would be to go to some reliable engine builder and ask him to build or specify an engine that would perform the service required. Having only one builder intrusted, the item of expense would not be chief in his consideration since there would be no competition, therefore the builder would build or specify the best engine possible for the service. If the funds available are limited or must be closely conserved, the intended purchaser may state the limits of cost and then require the builder to come within those limits. It would also be wise on the part of the purchaser to require a guarantee as to the performance of the engine and its maintenance cost for a given period of one year or more.

Drawing Up Specifications. If the purchaser is a competent engineer or he has in his employ such a person, a complete set of specifications may be drawn up and submitted to several engine builders for competitive bids. The specifications submitted should cover in detail the service for which the engine is to be used, the speed at which it is to operate, the type of valves and valve gear desired, the per cent of variation permissible in its governing, and many other items as to the design and detail of construction. Most specifications also specify within what limits the engine must operate, as to the amount of steam used per indicated horsepower per hour, and the range of mechanical efficiency that must be attained. A pro-

vision should be made in the contract as to the conditions under which the acceptance test will be made and by whom.

The form of specification usually submitted by the builders and which in general will be like those written by an engineer when requesting bids, is submitted herewith. This may be taken as a typical specification, the items being changed to meet different conditions of service as the particular case demands.

SPECIFICATIONS OF A VERTICAL CROSS-COMPOUND, SIDE-CRANK, ENGINE, ARRANGED FOR 1000-K.W. DIRECT CONNECTED GENERATOR, 60 CYCLE ALTERNATOR

Size, Power, and Dimensions

Diameter of high pressure cylinder, 27 inches.
Diameter of low pressure cylinder, 54 inches.
Stroke, 42 inches.
Revolutions per minute, 120.
Initial steam pressure, 125 pounds, 26 inches vacuum, condensing.
Rated load in indicated horsepower, 1,520; cut-off, 26/100.
At $\frac{1}{2}$ cut-off, indicated horsepower, 2,100; maximum cut-off, 7/10.
Estimated total weight of engine, 346,000 pounds.
Weight of wheel, 92,000 pounds. Diameter, 16 feet. Face, —— inches.
Diameter of bearings, 19 inches. Length, 35 inches.
Diameter of shaft between bearings, 22 inches.
Diameter of crank pin, 9 inches. Length, 8 inches.
Diameter of crosshead pin, 8 inches. Length, 8 inches.
Bearing surface of crosshead, 17 inches by 20 inches.
Diameter of piston rod, 5 inches.
Diameter of throttle valve, 12 inches.
Diameter of exhaust opening, 22 inches.

Workmanship and Materials

The workmanship, finish, fitting, and materials will be first-class in every particular. All forgings will be of open-hearth steel or hammered iron, as hereafter specified. All castings subject to wear, such as cylinders, guides, pistons, etc., will be poured from mixtures containing charcoal iron, graded according to the size of casting in order to secure the proper hardness and closeness of grain.

The engine will be made to gauge and interchangeable. This feature will be thoroughly carried out.

Flat surfaces will be scraped to surface plates, and surface and cylindrical grinding will be used where advantageous.

Guarantee

We guarantee the workmanship and materials in the engine to be first-class and in fulfillment of our guarantee we will give a duplicate to take the place of any part that may prove defective in material, workmanship, or design within one year after the engine is started.

We guarantee the engine to regulate from no load to full rated load within 2 per cent variation of speed.

We guarantee the engine to run in a smooth and proper manner without undue heating or vibration.

CYLINDERS

The cylinders and steam chests will be neatly covered with sheet iron lagging, enclosing a thick layer of the best quality of asbestos or magnesia fiber. The cylinder and steam chest covers will also be provided at each end with thin iron castings or covers. The cylinders will be provided at each end with a patent combination relief-valve and drip-cock of large diameter, adjustable to open automatically at any pressure desired. Being operated by hand as drip-cocks, these will not stick or become inoperative from disuse, but will relieve dangerous pressure from water or other causes.

JACKETS AND RECEIVER

The high pressure cylinder will be steam-jacketed and there will be a receiver of large capacity between cylinders.

The receiver will be filled with seamless brass heating coils containing steam at boiler pressure. The high pressure jacket and these coils should be piped in series, so that steam will pass through in the order named, and since the steam in the low pressure coils is hotter than the receiver steam, the latter will be considerably superheated upon entering the low pressure cylinder, and enough of the former will be condensed in the coils to cause brisk circulation in the high pressure cylinder jacket which is necessary to its efficiency. It is the aim of this arrangement to keep the steam dry throughout its course through the engine without the loss of any portion of heat of the jacket to the exhaust steam. The water condensed in jackets and in the coils should be returned to the boiler.

VALVES

Both cylinders will be four-ported and provided with valves of the flat gridiron type of our standard form.

The valves slide crosswise of the cylinder upon gridiron seats, which are separate and removable from the cylinder itself. Since the valves are of the gridiron type, a very small stroke is necessary to give full opening, and they move with an intermittent motion, standing still when closed, and only require power to operate when open and relieved of steam pressure. The clearance is reduced to about one-half of that necessary with valves of the Corliss type.

These valves possess the following advantages:

They give rapid opening of port with the least amount of wear and power required to operate.

The clearance space is reduced to a minimum.

They will not stick when the engine is started, and are easy to keep lubricated.

They wipe over and wear evenly, are unbalanced, and hence *will be tight* when old as well as when new.

VALVE GEAR

The main valves will be driven by a fixed eccentric controlling the admission of the steam and the opening and the closing of the exhaust. The cutting off of the steam will be effected by the cut-off valves which are controlled by the governor.

The valve gear is positive, composed of simple levers and links, and the cut-off can take place at any point between zero and the maximum cut-off. The cut-off, except at light loads, occurs when the main and cut-off valves are moving in opposite directions, and the cut-off is as sharp as with a releasing type of valve gear notwithstanding the short stroke used.

The cut-off is varied simultaneously upon all the cylinders in such a manner that the work done in each is approximately equal, as is also the drop in temperature of steam in each. This adds to smooth running and gives best distribution of steam for economy at all cut-offs under variable loads.

The valve gear will be constructed in the most substantial and durable manner, and in such a way as to equalize the cut-off at both ends of the cylinders for all cut-offs. Rock-shafts, pins, and links will be made of open-hearth steel. Connecting links will be fitted with bronze ends having quick taper key adjustment. The eccentric straps will be lined with babbitt hammered in and bored out. The rock-shaft bearing will be babbitted and adjustable.

Governors

The governor will be situated on the main shaft of the engine. A change in position of the centrifugal weights revolves the eccentric controlling the position and motion of the cut-off valves around the shaft and varies the point of cut-off.

All the bearing pins in the governor will be made of tool steel hardened and ground, turning in bearings bushed with phosphor bronze. The centrifugal force of each governor weight is resisted by a plate spring through a pin having hardened steel points resting in phosphor bronze cups, one at the end of the spring and the other at the center of gravity of the governor weight. The centrifugal force of the governor weights is thus opposed in a direct and frictionless manner without causing pressure or friction on the pins upon which the governor weights swing. This governor will regulate the speed of the engine with a closeness and certainty impossible with a fly-ball governor, and its action is unaffected by wide and sudden fluctuations of load. The governor will control both cut-off eccentrics.

Pistons, Piston Rods, and Stuffing Boxes

The pistons will be cored out and provided with internal ribbing, making them very light and strong. They will be secured to the piston rod by being forced upon a taper, with shoulder beyond, and by a nut, with a simple but efficient looking device. The pistons will be provided with cast-iron packing rings.

The piston rods will be of open-hearth steel running through deep stuffing boxes and babbitted glands. The rods will not touch the heads—which will be bored large—bronze rings fitting the rods in the bottom of each stuffing box and preventing escape of packing to the interior of the cylinder.

Low pressure piston will be of steel.

Framing

This will consist, for each cylinder, of a deep and massive base containing the main bearings. On the back of each base will stand a very heavy rectangular column, as shown in the blue print, securely bolted to a heavy frame head. In front the frame heads will be connected to the bases by forged steel columns bolted by flanges forged solid with the columns. The

rear column will support the cylinders when the forged columns in front are removed, facilitating the placing of shaft and other parts.

GUIDES, CROSSHEADS, AND CROSSHEAD PINS

The guides will be separate from the frame and adjustable for wear with an oil dish at the bottom which, together with a thin brass fringe upon the bottom of the crosshead, forms an efficient self-oiling device.

The crossheads will be of open-hearth steel fitted with babbitted cast-iron shoes.

The crosshead pins will be of open-hearth steel flattened on two sides to prevent wearing oval.

CONNECTING RODS AND BOXES

The connecting rod will be of forged steel, provided with gib and key ends. The straps will be provided with pinching bolts which will prevent spreading. Both crank and crosshead pin boxes will be lined with babbitt hammered in and bored out.

The body of the connecting rod will be made of larger section than the piston rod, being designed properly for the added strain due to its length and angular motion.

SHAFT, CRANK PIN, AND DISK

The shaft will be piled and faggoted hammered iron forging.

The crank disk will be made with counterbalance, of a mixture containing charcoal iron. The crank pin will be made of forged steel. The shaft and crank pin will be forced into the disk by hydraulic pressure and the disk will also be keyed securely to the shaft.

MAIN BEARINGS AND REMOVABLE SHELLS

The main bearings will be fitted with cylindrical shells, lined with babbitt, hammered in and bored out. These shells can easily be taken out by removing the cap and simply jacking up the shaft sufficiently to take the weight off the bearings, when they can be revolved around the shaft and taken out without disturbing any other parts of the engine. The shells are made hollow for water circulation. This is not intended to be used ordinarily, but in case dirt or other unusual conditions should cause the bearing to heat, it often enables the engine to complete its run without stopping.

The main bearings will be provided with a self-oiling device which will keep them flooded with oil.

OIL FEED SYSTEM

The feed will be positive and adjustable and the system will be closed, so that there will be little waste and deterioration of oil. Rings at the ends of the bearings will throw off escaping oil into close-fitting shields with suitable drain pipes leading to a large settling reservoir beneath. A small pump driven from the valve gear will deliver the oil to a feed tank at each bearing. This tank will be provided with an adjustable feed outlet pipe leading to the bearings, and with a gauge-glass and by-pass overflow, and can be filled by hand and used as an ordinary oil cup if it is desired to cut off the automatic supply while the engine is running.

FLYWHEEL

The wheel will be cast in halves and will be bolted together at the hub with reamed bolts carefully fitted in holes drilled from the solid, and the parts will be planed where they join. Steel arrow head links will be used

at the rim. The wheel will be carefully designed throughout in order to have a large factor of safety, and both edges and face of rim will be turned true.

PLATFORMS

Platforms convenient for handling and operating the engine will be provided as shown in print. These can be arranged to suit the location of the engine and will be made stiff to avoid vibration. The hand railings will be of seamless brass tubing, fitted into brass caps or iron posts. The platform plates will be diamond figured, planed where they join together and neatly fitted. Stairs will be made of channel iron, with cast-iron diamond threads.

FIXTURES

The following fixtures will be provided: throttle valve; indicator motion; complete outfit of sight-feed cylinder lubricators; glass body oil pumps; grease cups for valve gear; centrifugal crank pin oilers; reservoirs with sight-feed outlets; oil pipes and wipers for oiling the main parts of the engine conveniently and continuously; relief valves for each end of the cylinders; drip-cocks; wrenches, foundation bolts; and foundation plans.

Contract. After the engine has been selected and the builders determined, a written contract should be entered into in order to make it a legal document. A contract, according to Blackstone, is an agreement upon sufficient consideration to do or not to do a particular thing. In the case of the purchasing of an engine, the builder agrees to build, erect, and put into operation an engine in accordance with the specifications and drawings submitted, which items become a legal portion of the contract. The purchaser may also require that the engine be ready for operation in a given time and that it must also come up to certain requirements in its performance, as previously mentioned. In consideration of the foregoing, the purchaser agrees to pay the builders a specified sum of money, either in one payment or more as determined by them. The wording and statement of the contract should be carefully prepared, in order to avoid any possible misinterpretation of any of its provisions.

COST OF ENGINES AND OF THEIR OPERATION

The question of the cost of an engine and of its erection and operation is indeed a very vital one. This cost can not be classified in a brief way, since there are so many contributing factors that differ widely in different localities. For example, no well-defined indication of the cost of operation can be given, and the cost of labor and material are fluctuating items of expense; therefore, the cost of the engine can not be stated definitely, since in a brief interval of time it may be considerably more or less. Many articles appear

TABLE III

Price of Single Cylinder Corliss Engines, Set and Erected

Size of Cylinder, Inches	Horse-power	Cost of Engine	Cost of Foundation	Cost of Erecting	Cost of Piping	Total Cost
16×36	125	$1950	$325	$210	$180	$2665
18×36	155	2150	375	240	200	2965
18×48	200	2600	425	260	220	3505
20×48	230	2850	525	275	250	3900
22×42	250	3000	550	300	310	4160
24×48	320	4000	700	375	390	5465
28×48	425	5150	900	500	800	7650
30×48	490	5800	1200	600	1070	8670

from time to time in the leading engineering papers which give valuable information upon such matters and usually this information is correct since it is given currently with the ascertained cost of various items. It is, therefore, suggested that if the latest and perhaps most authentic information is desired upon these items of expense that such articles as appear in the papers mentioned should be consulted.

Engine Costs. As an indication of what such expense will be Tables III and IV, as devised by Dean C. H. Benjamin, are given.

TABLE IV

Cost of High Speed, Single Cylinder Engines

Horsepower	Size of Cylinders Inches	Steam Pressure Pounds Per Square Inch	R. P. M.	Cost Del. F. O. B.	Cost of Sub-Base	Cost of Engine Foundation	Cost of Superintendence—Labor	Cost of Handling	Total Cost (Engine set up on Foundation)
50	9×10	100	300	$695	$45	$65	$70	$10	$885
75	10×12	100	300	890	50	75	70	15	1100
100	12×12	100	290	1085	50	80	70	15	1300
125	13×14	100	275	1260	70	95	70	17	1512
150	15×14	100	245	1595	80	110	75	20	1880
	14×16								
200	18×16	100	225	2010	90	140	85	25	2350
250	19×18	100	200	2800	250	200	100	35	3385

Relative Cost of Operation Items. The cost of the operation of a steam plant is properly made up of several items, viz, rent or

TABLE V

Cost of Installation and Operation for One Year

Kind of Engine	Total Cost Engines and Boilers	Annual Cost of Both Engine and Boilers, Depreciation and Interest	Total Tons Coal Per Year of 3000 Hours	Lubricants	Labor, Engine and Boilers	Cost of Power, Coal at $2 Per Ton, 1 Year
Simple Slide Valve Non-condensing	$29.75	$4.03	6750	$1.02	$5.00	$23.55
Compound Slide Valve Non-condensing	31.50	4.38	5660	1.25	4.50	21.45
Compound Slide Valve Condensing	29.80	4.26	4050	1.25	3.80	17.41
Simple Corliss Non-condensing	32.25	3.84	6075	1.00	4.70	21.00
Compound Corliss Condensing	30.87	3.76	3375	1.25	3.50	16.25
Triple Corliss Condensing	34.25	4.28	3110	1.50	4.00	16.00

interest on real estate; interest on investment; maintenance, etc., of equipment; fuel; water; supplies; and attendance.

The relative value of these various items for a large central station lighting plant was given in the *Engineering Magazine*, May, 1905. Taking the total cost of maintaining the station as 100 per cent, the following were the average costs of the various items: Fuel 52.5%; wages 26.4%; water 2.2%; oil and waste 1.8%; rent 4.35%; station repairs 2.2%; steam repairs 5.45%; electric repairs 5.1%.

Annual Operation Expenses. Professor Carpenter in the *Economist* summarizes the cost of installation and the operation of an entire plant for one year of 3,000 hours as given in Table V. A coal consumption of 4.5 pounds per boiler horsepower per hour is assumed and the cost given per engine horsepower is for a 1,000 horsepower engine.

An illustration involving the items given in Table V will serve to make it clearer. The case of a simple slide valve non-condensing plant will be considered.

Cost of engines and boilers at $29.75 per horsepower = $29,750.
Annual cost of depreciation and interest at $4.03 per h.p. = $4,030.
Annual cost of coal at $2.00 per ton = 6750 × 2 = $13,500.
Annual cost of lubricants at $1.02 per horsepower = $1.02 × 100 = $1,020.
Annual cost of labor at $5.00 per horsepower = $5.00 × 1000 = $5,000.

Annual cost for the last four items = $23,550 or $23.55 per h.p.

The foregoing tables will serve to give some idea of the cost of engines, also of the cost of operation of a steam plant, but it must be remembered that the figures given will not be exact for all localities or for all times, due to the changing influences previously mentioned.

ENGINE TESTS

Importance of Tests. It was mentioned in connection with the discussion of specifications and contracts that often a guarantee is given by the builder as to the economical performance of a steam engine, hence it is required that the engine be tested in order to ascertain whether or not it meets the provisions of the guarantee. While this is one reason that may be assigned for testing an engine, yet there are several others of importance. The user from time to time may want to ascertain the condition of the engine as a whole and also the condition of particular features such as the valves, etc. For purely theoretical reasons an engine is often tested in order that an analytical study may be made of its performance under various conditions and in comparison with other engines of different classes. Many such tests have resulted in obtaining data, the facts of which have demonstrated to both the builder and the user possible economies. Because of the information thus obtained, the builder has been enabled to design a better engine, and the user to operate his engine more advantageously. The remarks given will suffice to indicate that the ultimate object of an engine test is the determination of the *economy* with which the engine produces a given amount of *power*. In steam engines the economy, as usually ascertained, relates to the weight of steam consumed, to the quantity of coal required to make the steam, or to the number of heat units supplied. The elementary quantities concerned are accordingly two in number, viz, the amount of steam, fuel, or heat (as the case may be) consumed, and the amount of power developed. How to determine these quantities is the problem.

A. S. M. E. Code. The American Society of Mechanical Engineers (A.S.M.E.) deemed the testing of engines according to some definite and standard method of such importance that a committee was appointed to devise a standard code. This, after much labor and diligent study, was presented to the Society and adopted. The full report appears in Volume 24 (1904), page 713, of the Trans-

actions. This code indicates the method of obtaining the required data and also gives a recommended form of report. In so far as the conditions will permit, this code should be followed. The several items of the code are summarized as follows:

Method of Conducting Steam Engine Tests—Code of 1902

(*1*) *Object of Test.* Ascertain at the outset the specific object of the test, whether it is to determine the fulfillment of a contract guarantee, to ascertain the highest economy obtainable, to ascertain the performance under special conditions, to determine the effect of changes in the conditions, or to find the performance of the entire boiler and engine plant, and prepare for the test accordingly.

No specific rules can be given for the preparations for the test as conditions surrounding each test will require a more or less different solution, which must be solved by the one in charge. The one particular thing to be emphasized at all times is, that in order to obtain data that is absolutely reliable for the purpose in view, the one in charge must be vigilant, conscientious, and, above all, honest.

(*2*) *General Condition of the Plant.* Examine the engine and the entire plant concerned in the test; note its general condition and any points of design, construction, or operation which bear on the objects in view. Make a special examination of the valve and pistons for leakage by applying the working pressure with the engine at rest and observe the quantity of steam, if any, blowing through per hour.

If the test is for the purpose of ascertaining the highest efficiency, the valves and pistons must be made steam- and water-tight.

To test the valves and piston block the flywheel so the piston will be near one end of the stroke, and turn on the steam. The leakage will escape to the exhaust port and will be observable. Another approximate method is to block the engine as before and having an indicator on the cylinder, observe the drop in pressure after an interval of time. In a tight engine the fall of pressure will be slow, whereas in a leaky one it will be fast. Other methods of determining the amount of leakage may suggest themselves to the engineer in charge of the test.

(*3*) *Dimensions, etc.* Measure or check the dimensions of the cylinders in every case, this being done when they are hot. If they are much worn, the average diameter should be determined. Measure also the clearance, which should be done, if possible, as described

in "Valve Gears." If the clearance can not be determined directly, it can be done approximately from the working drawings of the cylinder.

(4) *Coal.* When the trial involves the complete plant, embracing boilers as well as engine, determine the character of the coal to be used. The class, name of the mine, size, moisture, and quality of the coal should be stated in the report.

(5) *Calibration.* All instruments and apparatus should be calibrated and their reliability and accuracy verified by comparison with recognized standards. Such apparatus which is liable to change or become broken during the test, as gauges, indicator springs, and thermometers, should be calibrated before and after the test. The accuracy of scales should be verified by standard weights. Any water meters used should be carefully calibrated before and after, and during the test if possible. All of the apparatus may be very easily tested, directions for which may be found in various handbooks of engineering and often in instruction pamphlets published by the manufacturers. The method of testing indicator springs is given in "Steam Engine Indicators."

(6) *Precaution.* Make sure that there is no leakage at any of the connections with the apparatus provided for measuring and supplying the feed water and the steam that could affect the results.

(7) *Duration of Test.* The duration of a test should depend largely upon its character and the objects in view. The standard test of an engine and, likewise, a test for the simple determination of the feed water consumption should be continued for at least five hours, unless the class of service prevents a continuous run of so long duration.

When the water discharged from the surface condenser is measured for successive short intervals of time, and the rate is found to be uniform, the test may be of much shorter duration than when the feed water is measured to the boiler. The longer the test with a given set of conditions, the more accurate the work.

The commercial test of a complete plant, embracing boilers as well as engines, should continue at least one full day of twenty-four hours, whether the engine is in motion the entire time or not. A continuous coal test of a boiler and engine should be at least of ten hours duration, or the nearest multiple of the interval between times of cleaning fires.

(8) *Starting and Stopping a Test.* Before beginning to take readings the engine should be operated a sufficient length of time for

it to become thoroughly heated in all of its parts, and in the meantime all of the measuring apparatus should be properly adjusted. Having made the preliminary arrangements mentioned, at a given signal the height of water in the gauge glasses of boilers is observed, the depth of water in the reservoir from which the feed water is supplied is noted, the exact time of day is observed, and the test is held to commence. Thereafter the measurements determined upon for the test are begun and carried forward until its close. It is convenient to begin the test at some even hour or minute, but the important thing is to begin the test when accurate readings can be obtained irrespective of the time. When the time for closing the test arrives, the water in the glasses should be brought to the same height as it was at the beginning; if this is not possible, corrections must be made in the report.

(9) *Measurements of Heat Units Consumed by the Engine.* The measurement of the heat consumed requires the measurement of all the water supplied to the boiler, by whatever means; the temperature of the water supplied from each source; together with the pressure and quality of the steam, which are to be taken at some point near the throttle valve. The quantity of steam used by the steam calorimeter must also be accounted for.

The heat to be determined is that used by the entire engine equipment, embracing the cylinders and all auxiliary cylinders and mechanisms concerned in the operation of the engine, including air pumps, feed pumps, reheaters, etc.

(10) *Measurement of Feed Water, or Steam Consumption of Engine, etc.* The method of determining the steam consumption applicable to all plants is to measure all the feed water supplied to the boilers and deduct therefrom the water discharged by separators and drips, as also the water and steam which escapes on account of leakage of the boiler and its pipe connections and leakage of the main and branches connecting the boiler and engine. In plants where the engine exhausts into a surface condenser the steam consumption can be measured by determining the quantity of water discharged by the air pump, corrected for any leakage of the condenser and adding this to the steam used by jackets, reheaters, and auxiliaries as determined independently.

In measuring the water it is best to carry it through a tank or tanks resting upon platform weighing scales suitably arranged for the purpose, the water being afterwards emptied into a reservoir beneath, from which the pump is supplied.

(*11*) *Measurement of Steam Used by Auxiliaries.* Although the steam used by the auxiliaries was measured as mentioned in item (*10*), yet it is very desirable to ascertain the steam consumption of each auxiliary independently, in order that a close analysis of the engine performance can be made. Several means may be employed for determining the steam consumption of the various auxiliaries, and since they will be apparent to the operator no discussion of them will be given, but they are only mentioned in order to emphasize the desirability of obtaining such data.

(*12*) *Coal Measurement.* In commercial tests of the combined engine and boiler equipment, or those made under ordinary conditions of commercial service, the test should extend over the entire period of the day, that is, twenty-four hours, or a number of days of that duration. Consequently, the coal consumption should be determined for the entire time. If the engine runs but a part of the time and during the remaining portions the fire is banked, the measurement of coal should include that used for banking. It is well, however, in such cases to determine separately the amount consumed during the time the engine is in operation and that consumed during the period while the fires are banked, so as to have complete data for purposes of analysis and comparison, using suitable precautions to obtain reliable measurements. The measurement of coal begins with the first firing, after cleaning the furnaces and burning down at the beginning of the test, and ends with the last firing, at the expiration of the allotted time.

In connection with coal measurements, whatever the class of tests, it is important to ascertain the percentage of moisture in the coal, the weight of ashes and refuse, and, where possible, the approximate and ultimate analysis of the coal. (For discussion of this item of coal the student is referred to Vol. 22, P. 34, of the A. S. M. E. Transactions.)

(*13*) *Indicated Horsepower.* The indicated horsepower should be determined from the average mean effective pressure of diagrams taken at intervals of twenty minutes, and at more frequent intervals if the nature of the test makes this necessary for each end of each cylinder.

The indicator diagrams should be taken at regular intervals but not necessarily simultaneously at the two ends of the cylinder. If the diagrams vary so much as not to give fair results, the diagrams should be taken more frequently.

The method of attaching, operating, and adjusting the indicator

and also the method to follow in obtaining the mean effective pressure are described in "Steam Engine Indicators."

(*14*) and (*15*) *Testing Indicator Springs and Brake Horsepower.* These items are fully discussed and explained in "Steam Engine Indicators."

(*16*) *Quality of Steam.* When ordinary saturated steam is used, its quality should be obtained by the use of a throttling calorimeter attached to the main steam pipe near the throttle valve. When the steam is superheated, the amount of superheating should be found by the use of a thermometer placed in a *mercury well* inserted in the pipe.

(*17*) *Speed.* There are several means for obtaining the number of revolutions the engine makes per minute. They may be counted during one minute or some other division of time, a tachometer may be used, but the most reliable results are obtained by using a revolution counter, such as was illustrated and described in "Steam Engine Indicators." In using the counter, the total reading should be taken each time the general test data is recorded. These revolutions per minute corresponding to the difference in reading of the instrument can then be computed, knowing the time interval.

(*18*) *Recording Data.* Take note of every event connected with the progress of the trial, whether it seems at the time to be important or unimportant. Record the time of every event, and time of taking every weight, and every observation. Observe the pressures, temperatures, water heights, speeds, etc., every twenty or thirty minutes when the conditions are practically uniform, and at much more frequent intervals if the conditions vary. Observations which concern the feed water measurements should be made with special care at the expiration of each hour of the trial, so as to divide the tests into hourly periods and show the uniformity of the conditions and tests as the test goes forward. Where the water discharged from the surface condenser is weighed, it may be advisable to divide the test by these means into periods of less than one hour.

The data and observations of the test should be kept on properly prepared blanks or in notebooks containing columns suitably arranged for a clear record.

(*19*) *Uniformity of Conditions.* In a test having for an object the determination of the maximum economy obtainable from an engine, or where it is desired to ascertain with special accuracy the effect of predetermined conditions of operation, it is important that all the conditions under which the engine is operated should be main-

tained uniformly constant. This requirement applies especially to the pressure, the speed, the load, the rate of feeding, the various supplies of water, the height of the water in the gauge glasses, and the depth of the water in the feed water reservoir.

(*20*) *Analysis of Indicator Diagrams.* (a) Steam accounted for by the indicator. The method of accounting for the steam by use of the indicator is thoroughly treated in "Steam Engine Indicators," so no further discussion will be given here. (b) Sample indicator diagrams. In order that the report of a test may afford complete information regarding the conditions of the test, sample indicator diagrams should be selected from those taken and copies appended to the tables of results.

The points at which the different events occur should be clearly marked on the cards submitted with the report.

(*21*) *Standards of Economy and Efficiency.* The hourly consumption of heat determined by employing the actual temperature of the feed water to the boiler, as pointed out in item (*9*), divided by the indicated and brake horsepower, that is, the number of heat units consumed per indicated or brake horsepower per hour—these are the standards of engine efficiency recommended.

It is useful in this connection to express the efficiency in its more scientific form, or what is called the "thermal efficiency ratio." The thermal efficiency ratio is the proportion which the heat equivalent of the power developed bears to the total amount of heat actually consumed, as determined by test. The heat converted into work represented by one horsepower is 1,980,000 foot pounds per hour, and this divided by 778 equals 2,545 British Thermal Units. Consequently the thermal efficiency ratio is expressed by the fraction

$$\frac{2545}{\text{British Thermal Units per hour}}.$$

(*22*, *23*, *24*, and *25*). These sections of the Code deal with purely scientific investigations, hence they do not essentially enter into commercial tests and will not be given.

(*26*) *Report of Test.* The data and results of the test should be reported in the manner and in the order outlined in the following report. It is the intention that the report be full enough to apply to any type of engine, but when not so, or where special data and results are determined, additional results may be inserted under the appropriate headings.

Actual Engine Test. To illustrate the application of many of the items given as obtained from the Code, a full engine test will be

taken and reported upon. This report will serve to give the order and manner in which data should be tabulated and also the method in which the report should be worked up.

DETERMINATION OF EFFICIENCY OF A BUCKEYE ENGINE UNDER DIFFERENT LOADS

Purpose

The purpose of this series of tests on the Buckeye engine located in the Engineering Laboratory of Purdue University was to determine the best efficiency under six different loads, ranging from zero to $1\frac{1}{4}$ load, by $\frac{1}{4}$ load steps, the engine running non-condensing and using 160 pounds of steam pressure, absolute.

Plan

The zero load was determined with the friction brake *I*, Fig. 70, removed and the engine running free. The full load was determined by the brake load

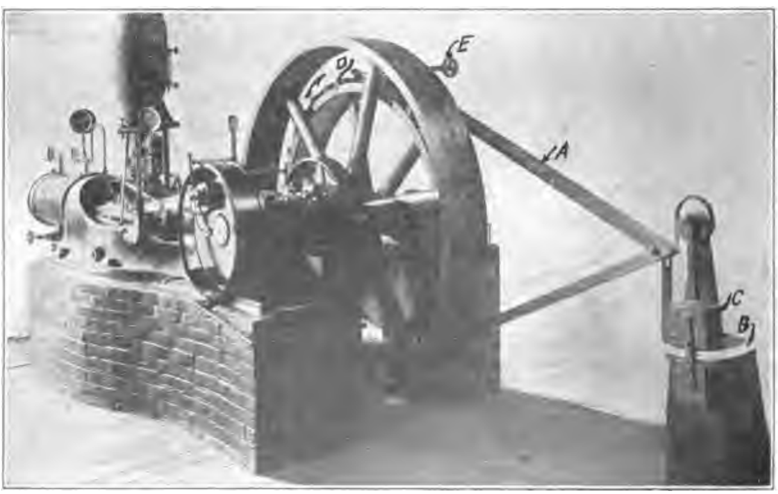


Fig. 70. Buckeye Engine Fitted with Prony Brake and Indicators

which the engine carried with 25 per cent cut-off, this being the builders' rating for this type of engine. The $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, 1, and $1\frac{1}{4}$ loads were taken as 25%, 50%, 75%, 100%, and 125%, respectively, of the full load.

Steam pressure was maintained constant at the pressure indicated for the test. Each test was of one hour duration, the engine having been run under conditions of the test a length of time sufficient to permit the conditions to become constant.

Method of Conducting Test

Constant steam pressure was obtained by throttling the 5-inch steam line leading to the engine by means of the pipe line valve. This throttling action was not sufficient to cause the steam to become superheated.

The revolutions per minute were obtained by means of a revolution counter.

Indicator diagrams were taken every five minutes, 13 sets of diagrams being obtained for each hour's run.

Barometer readings were taken every 15 minutes.

The amount of water was determined by condensing the exhaust steam at atmospheric pressure.

Preliminary Work

Before commencing the work the engine was placed in as good condition as was possible. The governor was adjusted in order to reduce friction; play was taken up in the valve gear and the valves were carefully set to give equal cut-off on both ends at full load; all stuffing boxes were repacked; the brake wheel was turned up and brake recalibrated.

The pressure in the engine supply line was obtained by tapping a $\frac{1}{4}$-inch pipe into the main, about 3 feet from the valve. This $\frac{1}{4}$-inch pipe was connected to a large steam gauge which faced the operator of the throttling valve, thus enabling him to watch the gauge all the time and maintain a constant pressure.

Observed Data

In each test the following observations were taken:

Steam pressure, constant throughout
Brake load
Revolutions per minute
Weight of condensed steam
Barometer
Indicator diagrams

Results

Having the above data it becomes possible to calculate the following:

(1) Per cent of cut-off, head end and crank end.
(2) Mean effective pressure (m.e.p.), head end and crank end.
(3) Indicated horsepower, head end and crank end and total.
(4) Brake horsepower (b.h.p.).
(5) Friction horsepower (f.h.p.).
(6) Mechanical efficiency.
(7) Pounds steam, per indicated horsepower per hour and per brake horsepower per hour.
(8) British Thermal Units per hour, per indicated horsepower and brake horsepower per hour.
(9) Thermal efficiency.

Constants and Formulas. The constants of the engine and formulas employed in obtaining the calculated items in the summary of results, are as follows:

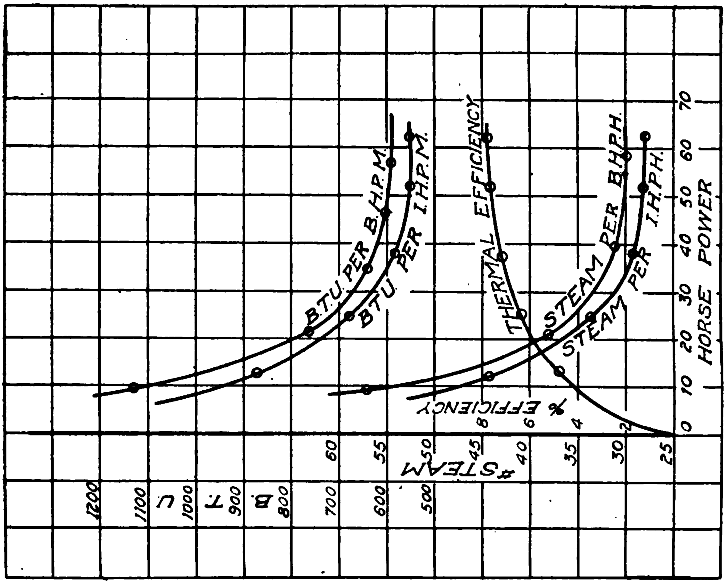


Fig. 72 Steam Consumption, B.T.U. and Thermal Efficiency Curves for Buckeye Engine

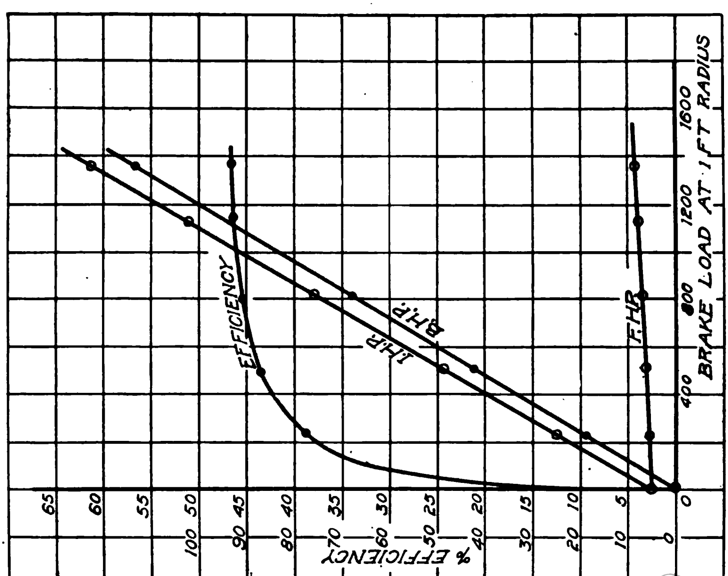


Fig. 71. Horsepower and Efficiency Curves for Buckeye Engine

Diameter of cylinder, 7.75 inches.
Piston rod diameter, 1.437 inches.
H.E. area, 47.173 square inches; c.e. area, 45.55 square inches.
Radius of brake arm, 38.25 inches, equals 3.185 feet.
Clearance, head end 6.15%; crank end, 6.765%.
Normal speed, 220 revolutions per minute.
Heat value of 1 horsepower, 42.42 British Thermal Units.
Heat value of 1 pound of steam, above 32°F. for 160 pounds absolute, 1,192.8 British Thermal Units.
Gauge pressure 15 pounds (approximate) less than absolute pressure.

The horsepower constants are as follows:

H.E.—i.h.p. Constant = .001787. (See "Steam Engine Indicators.")
C.E.—i.h.p. Constant = .001726.
B.H.P. Constant = .00060695.

Item (3). At observed revolutions per minute (r.p.m.):

H.E.—i.h.p. = .001787 × h.e. m.e.p. × r.p.m.
C.E.—i.h.p. = .001726 × c.e. m.e.p. × r.p.m.
Total i.h.p. = h.e. i.h.p. + c.e. i.h.p.

It often happens that the engine is not operated at the desired speed just at the instant of taking the reading, hence a correction must be made if the indicated horsepower is to be expressed and recorded for the normal speed. Therefore i.h.p. = total i.h.p. × 220 ÷ observed r.p.m.

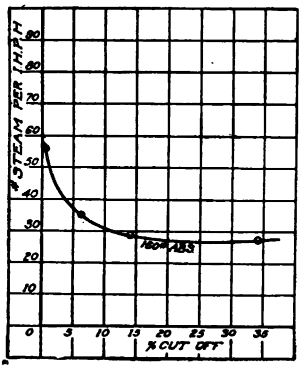


Fig. 73. Steam Consumption for Different Cut-Offs

Item (4). At observed r.p.m. determined as follows:

B.H.P. = .00060695 × pounds brake load × r.p.m.
B.H.P. = .0001904 pounds brake load at 1 foot radius × r.p.m.
At 220 r.p.m., corrected b.h.p. = b.h.p. × 220 ÷ observed r.p.m.

Item (5). At 220 r.p.m., the f.h.p. = total i.h.p − b.h.p.

Item (6). At 220 r.p.m., the mechanical efficiency = b.h.p. ÷ total i.h.p.

The pounds of steam per hour at 220 r.p.m. = pounds of steam per hour at observed r.p.m. × 220 ÷ observed r.p.m. The B.T.U. supplied per hour = corrected pounds of steam per hour × total British Thermal Units in 1 pound steam, at given absolute pressure above 32°F.

Item (7). The pounds steam per i.h.p. per hour = corrected pounds steam per hour divided by corrected i.h.p.

The pounds of steam per b.h.p. per hour = corrected pounds steam per hour divided by corrected b.h.p.

Item (8). The British Thermal Units per i.h.p per hour = total British Thermal Units supplied divided by 160 × corrected i.h.p.

The British Thermal Units per b.h.p., per hour = total British Thermal Units supplied divided by corrected b.h.p. × 160.

Item (9). The Thermal Efficiency = 42.42 British Thermal Units divided by British Thermal Units per i.h.p. per hour

TABLE VI

Indicator Diagram Data for Buckeye Engine Test

STEAM PRESSURE 160 # ABS.							
CYL. END	CARD NO.	NO LOAD		1/4 LOAD		1/2 LOAD	
		%C.O.	M.E.P.	%C.O.	M.E.P.	%C.O.	M.E.P.
HEAD END	1		.000	1.58	14.73	5.00	28.41
	2		.785	1.32	13.98	5.52	28.17
	3		.773	1.05	14.14	4.98	28.37
	4		.259	1.05	14.73	4.71	26.70
	5		1.040	1.05	13.65	5.00	26.82
	6		.000	1.05	14.45	4.46	26.78
	7		.000	.79	14.13	4.47	26.30
	8		.521	.79	13.41	4.71	26.18
	9		.675	1.05	12.90	4.71	27.22
	10		1.295	1.05	12.63	4.97	27.22
	11		.675	1.05	12.90	4.71	27.22
	12		.529	1.06	14.80	4.73	26.30
	13		.779	1.06	12.44	4.97	27.22
	AV.		.5638	1.073	13.757	4.841	27.14
CRANK END	1		4.325	.77	18.58	7.41	36.59
	2		4.325	.76	1850	7.66	36.80
	3		3.805	.77	18.62	7.65	36.46
	4		3.560	.77	18.58	7.65	36.20
	5		4.055	.77	19.13	7.65	36.20
	6		3.567	.77	18.92	7.66	36.30
	7		4.340	.77	18.43	7.65	36.20
	8		3.805	.77	18.37	7.41	36.30
	9		3.805	.76	17.70	7.64	37.63
	10		3.785	.76	17.95	7.41	37.81
	11		3.805	.76	17.95	7.65	36.75
	12		3.560	.76	17.75	7.40	36.20
	13		4.555	.76	17.95	7.69	37.18
	AV.		3.9445	.7653	18.338	7.578	36.655

TABLE VII

Indicator Diagram Data for Buckeye Engine Test

Steam Pressure 160 # Abs.							
CYL. END	CARD NO.	3/4 LOAD		FULL LOAD		1 1/4 LOAD	
		%C.O.	M.E.P.	%C.O.	M.E.P.	%C.O.	M.E.P.
HEAD END	1	13.85	46.70	24.90	63.25	33.92	77.60
	2	13.85	44.95	24.05	63.15	35.22	77.60
	3	13.69	44.75	24.85	62.85	35.32	77.80
	4	13.88	44.75	25.15	63.35	34.90	77.00
	5	14.40	44.50	24.52	64.65	35.41	76.60
	6	14.21	45.50	25.15	64.15	35.26	77.10
	7	13.85	44.65	24.21	64.20	34.27	77.55
	8	13.57	45.15	25.05	63.76	34.87	77.05
	9	13.65	44.90	25.15	64.90	36.21	78.95
	10	14.15	46.20	24.35	63.35	34.05	77.50
	11	14.70	45.40	24.42	63.25	34.07	77.95
	12	13.92	44.40	25.15	64.15	34.80	78.05
	13	14.11	44.15	25.20	64.50	34.72	77.95
	AV.	13.98	45.073	24.78	63.80	34.84	77.537
CRANK END	1	14.05	53.15	24.80	67.85	31.87	82.05
	2	14.50	52.80	24.75	67.30	34.25	82.30
	3	14.05	52.15	25.10	68.70	33.59	81.80
	4	14.32	53.45	24.05	67.50	32.59	80.80
	5	14.54	51.80	24.67	67.50	33.32	81.31
	6	14.11	53.10	24.80	68.35	32.83	81.30
	7	14.22	52.55	24.28	68.40	32.83	80.80
	8	14.54	52.25	24.61	68.00	32.83	81.00
	9	14.28	51.80	24.80	68.80	34.25	81.80
	10	14.50	51.85	24.38	69.06	32.90	80.25
	11	14.54	52.25	24.60	68.65	33.07	80.75
	12	14.54	53.30	24.58	68.55	33.52	80.00
	13	14.76	52.15	24.90	68.50	32.63	80.20
	AV.	14.38	52.527	24.59	68.30	33.11	81.10

TABLE VIII

PERFORMANCE OF
UNDER DIFFERENT CUT-OFF'S
SUMMARY

OBSERVED											
						AV. M.E.P.		I. H.P.			
LOAD	*BRAKE LOAD AT 1 FOOT RADIUS	R.P.M.	BAROMETER INS. OF MERCURY	*STEAM CONDENSED PER HOUR	PER CENT CUT-OFF	HEAD END	CRANK END	HEAD END	CRANK END	TOTAL AT OBSERVED R.P.M.	TOTAL CORRECTED TO 220 R.P.M.
0	0	287.1	29.82	434.5		.564	3.945	.289	1.953	2.242	1.72
¼	227.5	225.5	29.83	550.0	.97	13.76	18.39	5.45	7.13	12.66	12.35
½	531.0	222.9	29.81	849.0	6.21	27.14	36.66	10.82	14.10	24.92	24.58
¾	815.0	221.9	29.80	1096.0	14.2	45.07	52.53	17.86	20.10	37.96	37.62
1	1126.0	217.4	29.78	1401.0	24.6	63.80	67.74	24.80	25.62	50.42	51.00
1¼	1363.0	209.7	29.79	1614.5	34.0	77.54	81.10	29.00	29.30	58.30	61.20

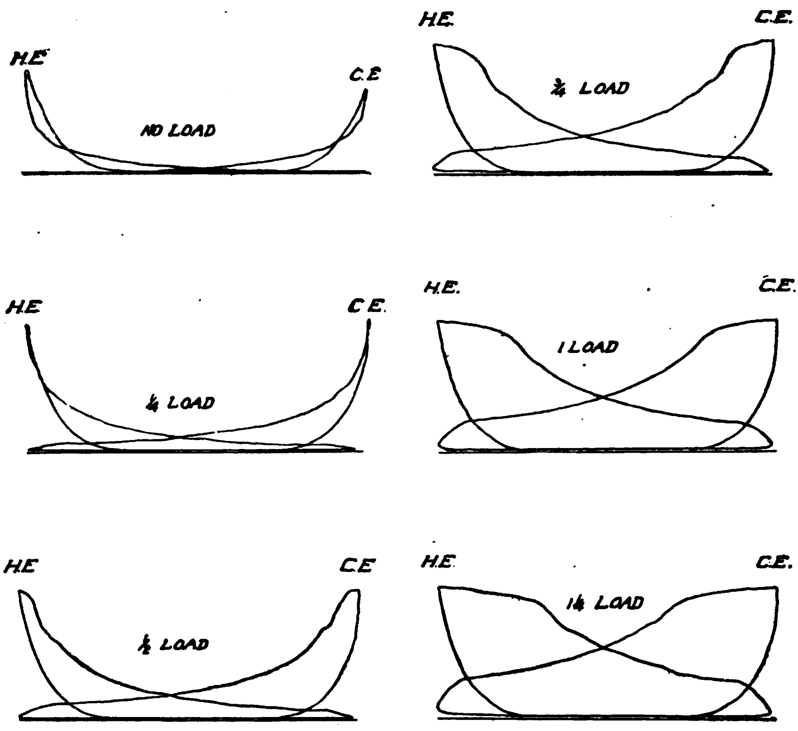


Fig. 74. Indicator Diagrams Taken During Test of Buckeye Engine

TABLE VIII—Continued

THE BUCKEYE ENGINE
STEAM PRESSURES 160# ABS.
OF RESULTS.

CALCULATED

B.H.P. AT OBSERVED R.P.M.	B.H.P. CORRECTED TO 220 R.P.M.	F.H.P. AT 220 R.P.M.	MECHANICAL EFFICIENCY	*STEAM PER HR. CORRECTED TO 220 R.P.M.	TOTAL B.T.U. SUPPLIED PER HOUR	*STEAM PER I.H.P.H.	*STEAM PER B.H.P.H.	B.T.U. PER I.H.P.M.	B.T.U. PER B.H.P.M.	THERMAL EFFICIENCY
0	0	2.35	0	332.5	396600	141.4		2810		0
9.75	9.54	2.71	77.2	547.0	652500	44.25	57.4	879	1140	4.82
21.81	21.51	3.07	87.5	837.0	998350	34.1	38.95	678	774	6.29
34.41	34.18	3.44	90.8	1085.0	1294300	28.85	31.80	573	632	7.25
46.55	47.17	3.88	92.5	1418.0	1691500	27.8	30.5	552.5	606	7.68
54.45	57.07	4.13	93.3	1695.0	2022000	27.75	29.7	551	590	7.70

Plotted Results. On curve sheet shown as Fig. 71, are plotted to pounds brake load at 1′ radius, the i.h.p., b.h.p., f.h.p., and mechanical efficiency.

On curve sheet shown as Fig. 72, are plotted to horsepower the pounds steam per i.h.p. per hour, the pounds steam per b.h.p. per hour, the British Thermal Unit per i.h.p. per minute, the British Thermal Unit per b.h.p. per minute, and the thermal efficiency.

On curve sheet shown as Fig. 73, is plotted a curve which shows the steam consumption for the different per cents of cut-off.

Conclusions and Comparisons

An examination of the curves shows a marked increase in economy of the ½ load over the ¼ load; a smaller increase in economy of the ¾ load over the ½ load; and a still smaller increase in economy of the full load over the ¾ load; but the full load and 1¼ load have the same steam consumption per i.h.p. per hour indicating that the engine is operating most economically throughout this range.

The tests indicate a very good range of economical operation from ¾ load to 1¼ load, and although the steam consumption is higher than the best recorded results for other engines of greater horsepower, yet the results obtained are very good considering the relatively small size of the engine.

Appendix

The engine under test was a 7¾″×15″ type "B" Buckeye engine which had been rebuilt from the old type of flat valve to a piston valve engine. The following information was supplied by the Buckeye Engine Company:

Lap 11/16″, Lead 1/16″, Compression 2½″, Exhaust laps, ⅛ and 1/16″, Clearance 5.6%, Cut-off 25%. Weight of reciprocating parts 150 pounds.

The arrangement of the brake apparatus may be seen in Fig. 70 in which *A* is brake lever, *B* is calibrated brake load arc, *C* is brake pendulum, and *D* is brake wheel.

Cooling water for the brake enters through a hose not shown in the illustration. The direction of rotation of the brake wheel is indicated by the arrow near *D*. By means of the hand wheel *E*, the brake load is applied and regulated. The brake was carefully calibrated before beginning the test.

Calibration of Constants

$$\text{H.E. piston displacement} = \frac{47.173 \times 15}{144 \times 12} = .41 \text{ cu. ft.}$$

$$\text{C.E. piston displacement} = \frac{45.55 \times 15}{144 \times 12} = .396 \text{ cu. ft.}$$

$$\text{H.E. clearance} = \frac{.0252}{.41} = 6.15\%$$

$$\text{C.E. clearance} = \frac{.0268}{.396} = 6.765\%$$

$$\text{H.E.} - \text{i.h.p.} \quad \text{Constant} = \frac{15 \times 47.173}{12 \times 33000} = .001787$$

$$\text{C.E.} - \text{i.h.p.} \quad \text{Constant} = \frac{15 \times 45.55}{12 \times 33000} = .001726$$

B.H.P. Constant = .00060695

$$\text{Thermal Efficiency Constant} = \frac{33000}{778} = 42.42$$

Tables VI and VII contain information from the indicator diagrams, and Table VIII is a general summary of the observed and calculated results of the tests.

Fig. 74 shows sample indicator diagrams taken during the test.

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